US006390869B2

(12) **United States Patent**
Korenjak et al.

(10) **Patent No.: US 6,390,869 B2**

(45) **Date of Patent: May 21, 2002**

(54) **FOUR STROKE ENGINE WITH VALVE TRAIN ARRANGEMENT**

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(73) Assignee: **Bombardier-Rotax GmbH**, Gunskirchen (AT)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/794,240**

(22) Filed: **Feb. 28, 2001**

Related U.S. Application Data

(60) Provisional application No. 60/185,703, filed on Feb. 29, 2000, and provisional application No. 60/257,174, filed on Dec. 22, 2000.

(51) **Int. Cl.**[7] **B63H 21/10**

(52) **U.S. Cl.** **440/88**; 123/90.23

(58) **Field of Search** .... 440/38, 88, 89, 440/84, 900; 123/73 AD, 317, 432, 90.23, 90.44, 196 W

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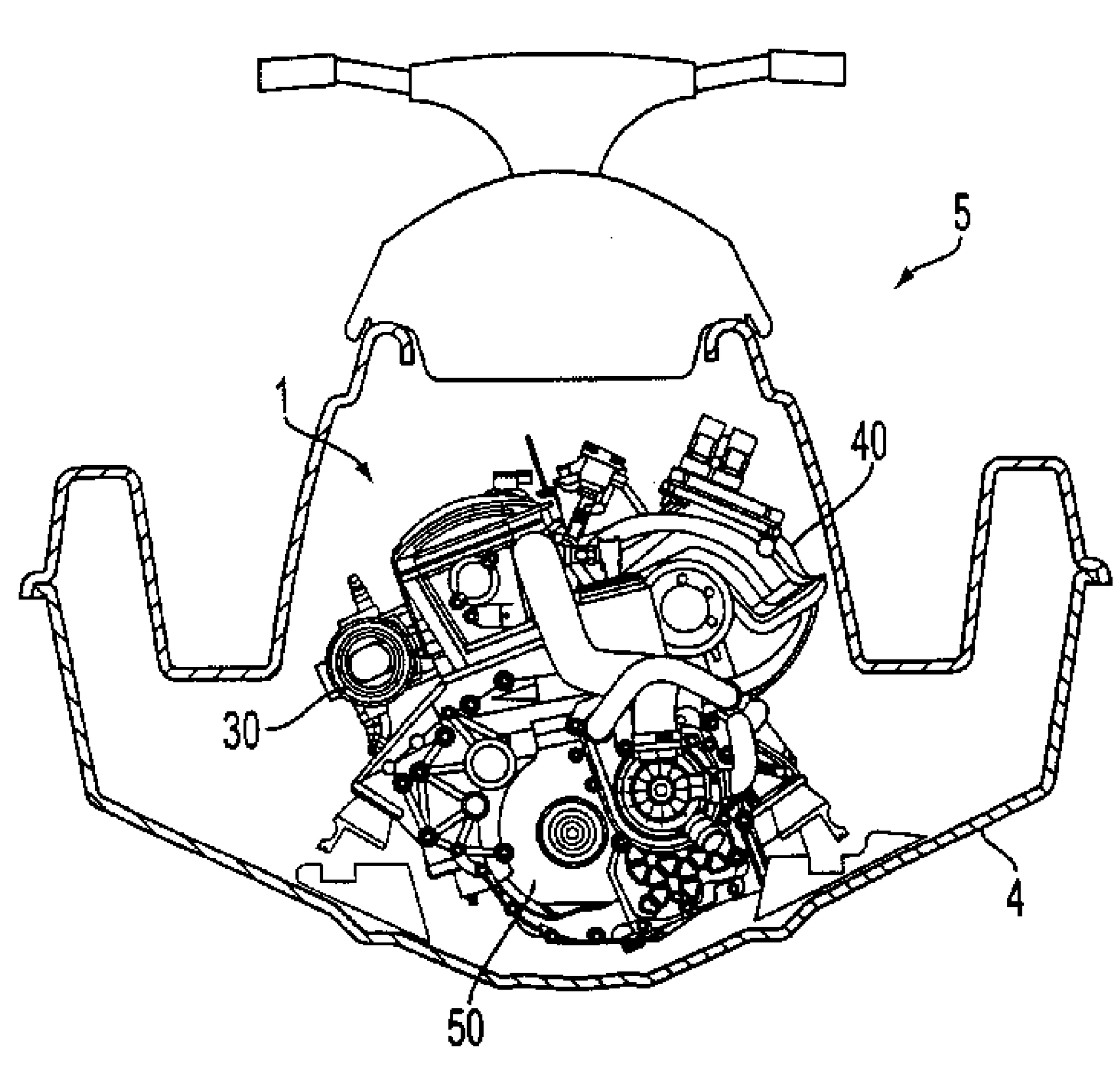
Primary Examiner—Stephen Avila

(74) *Attorney, Agent, or Firm*—Pillsbury Winthrop LLP

(57) **ABSTRACT**

A four stroke internal combustion engine includes a cylinder head having a valve train having reduced profile that permits access to a spark plug assembly located therein.

46 Claims, 42 Drawing Sheets



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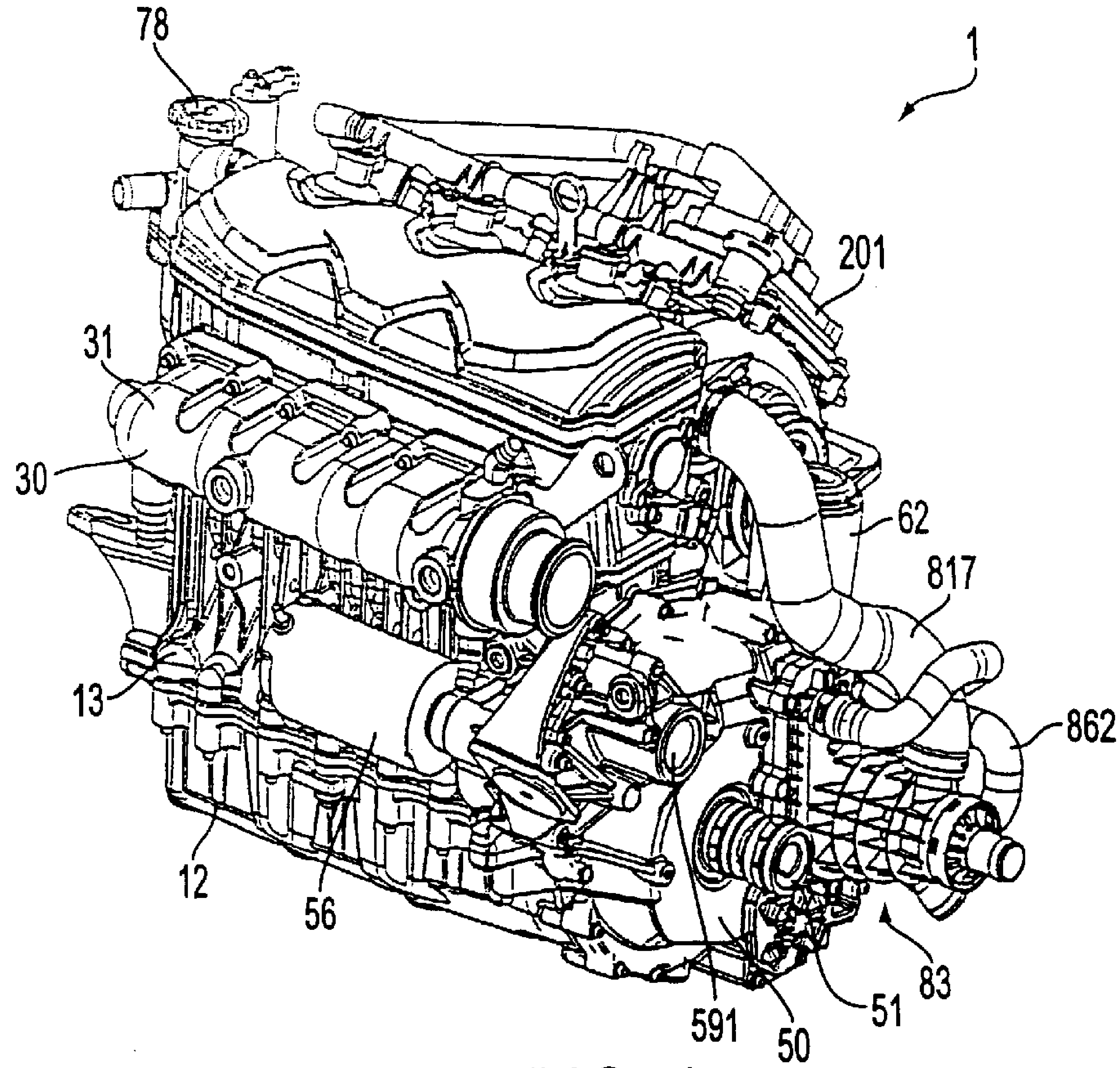
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FIG. 1

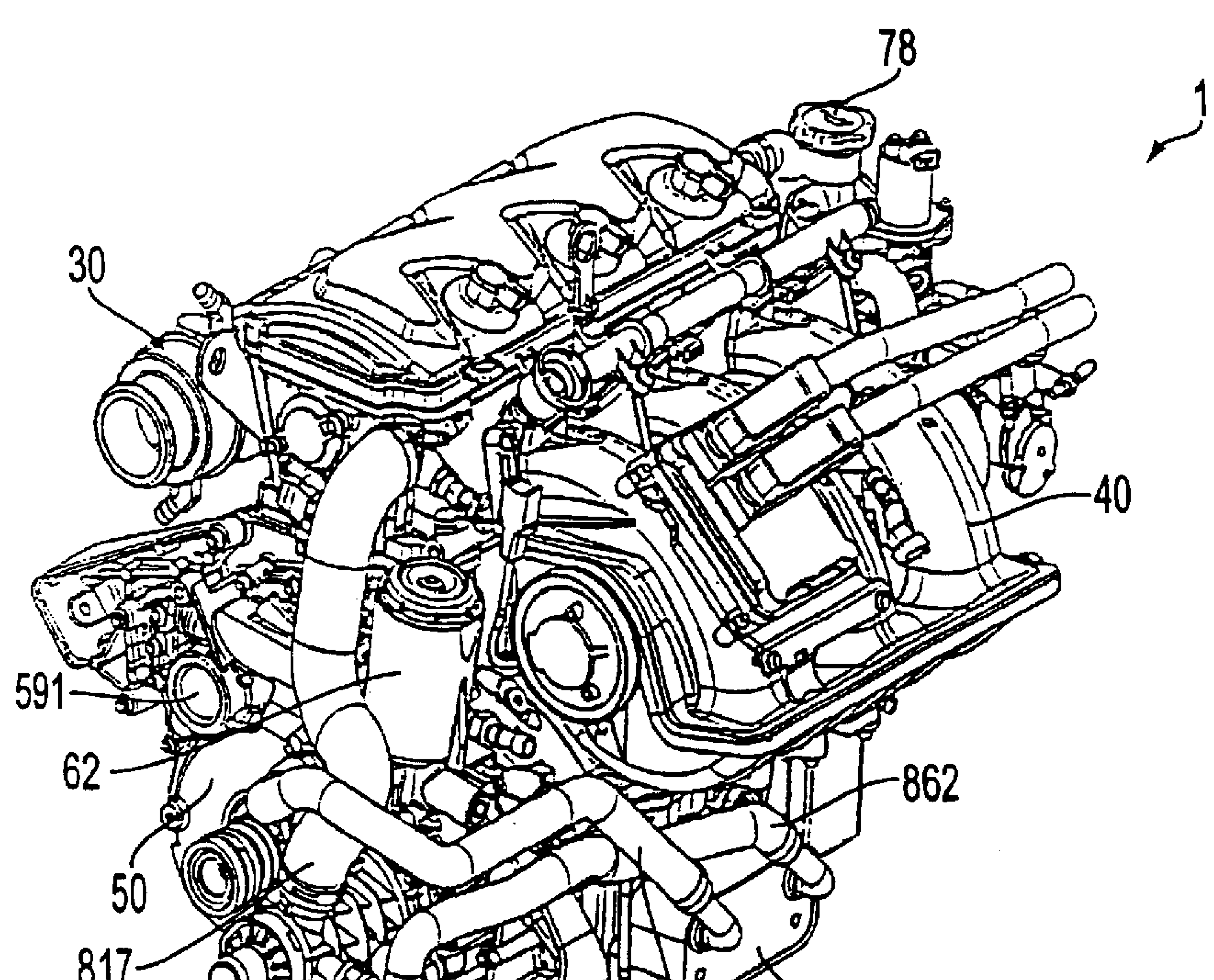
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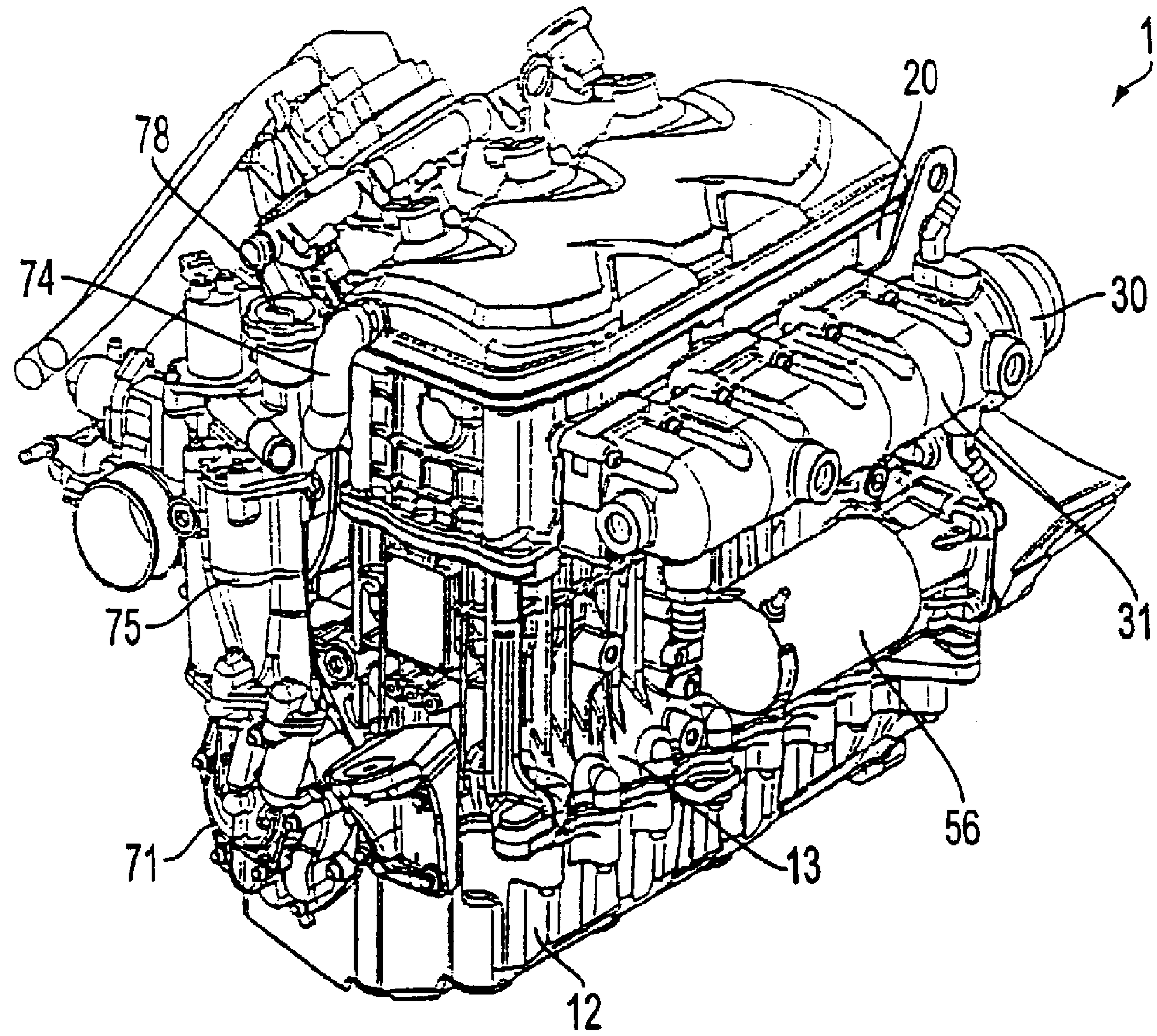
FIG. 2



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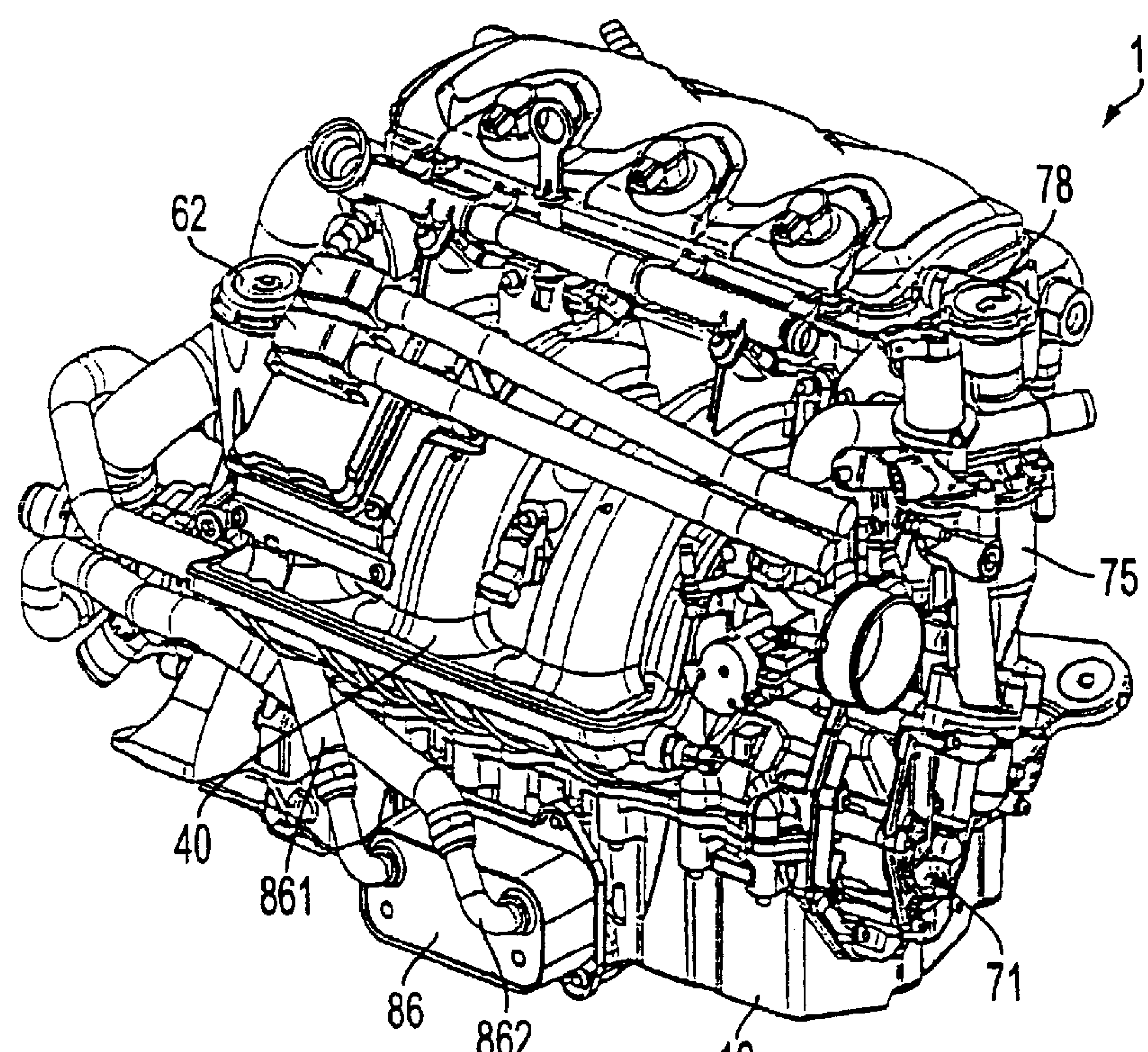

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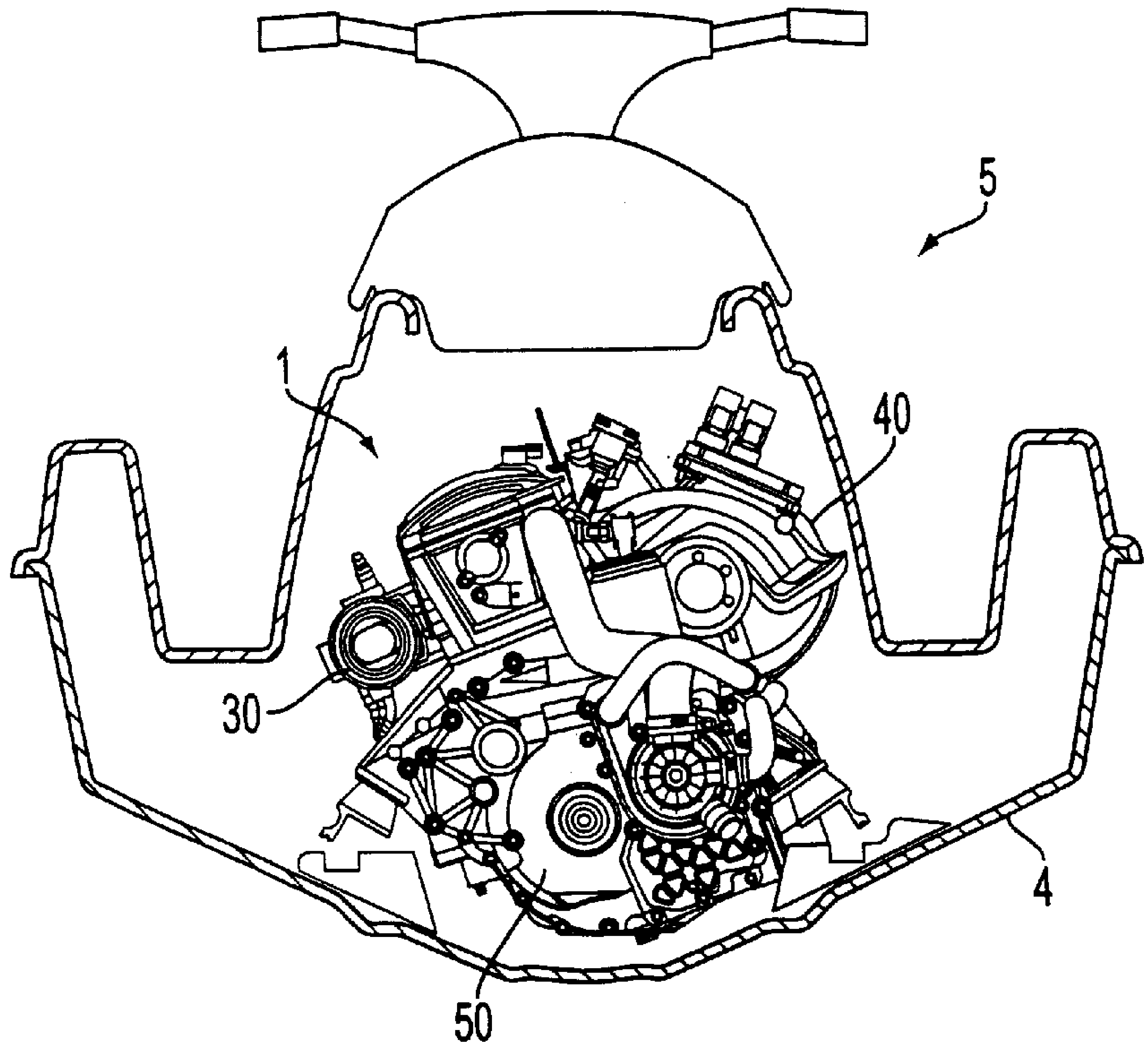

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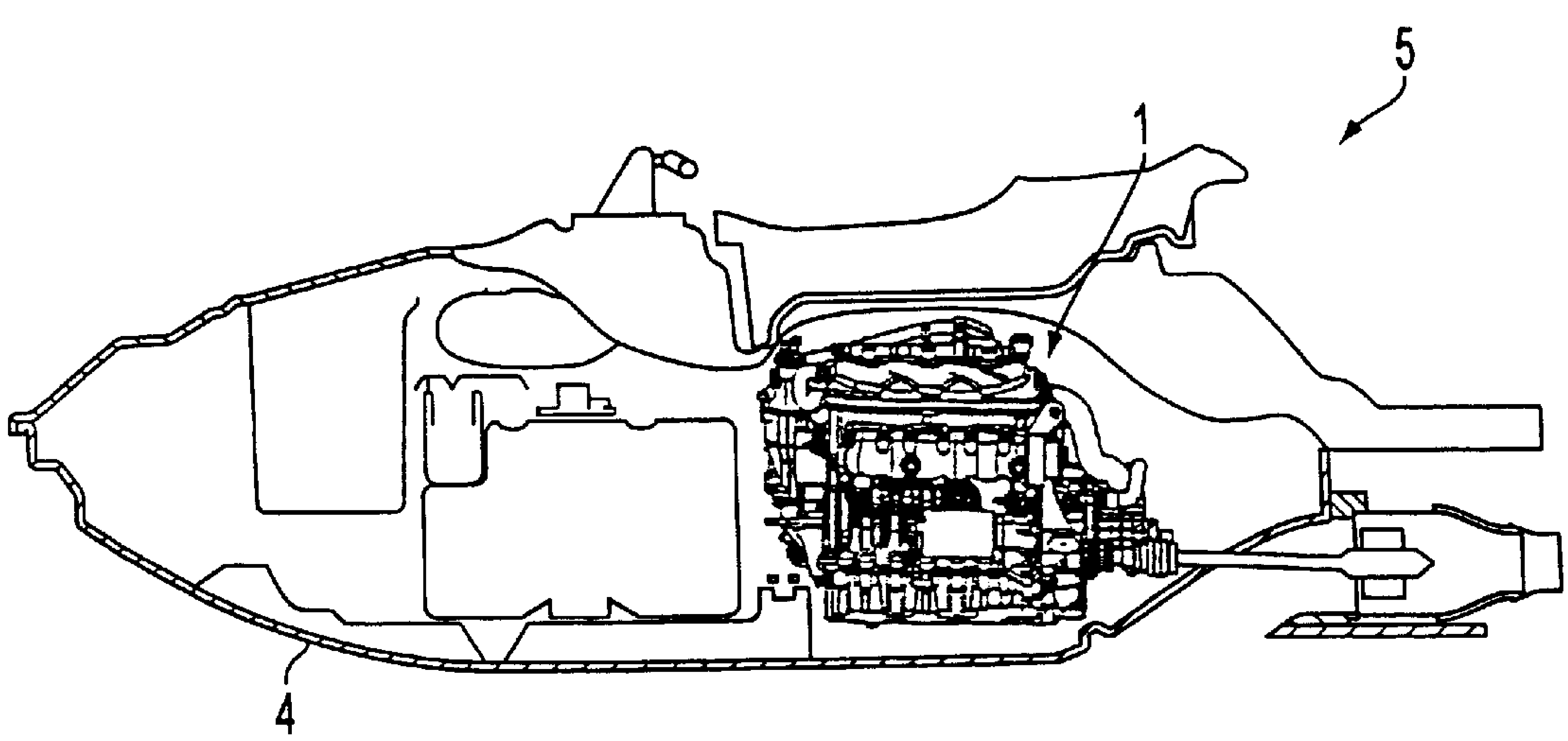

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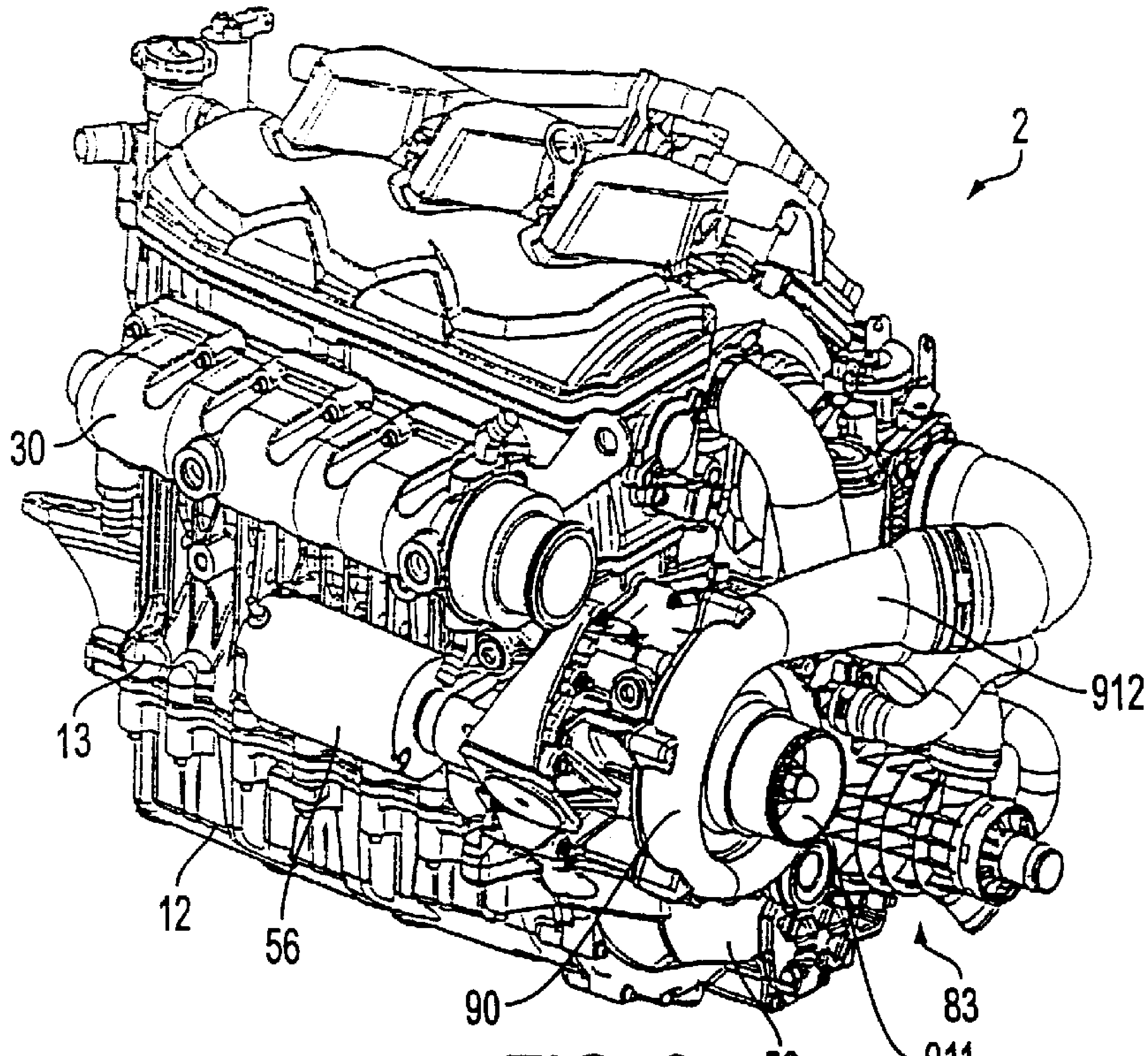

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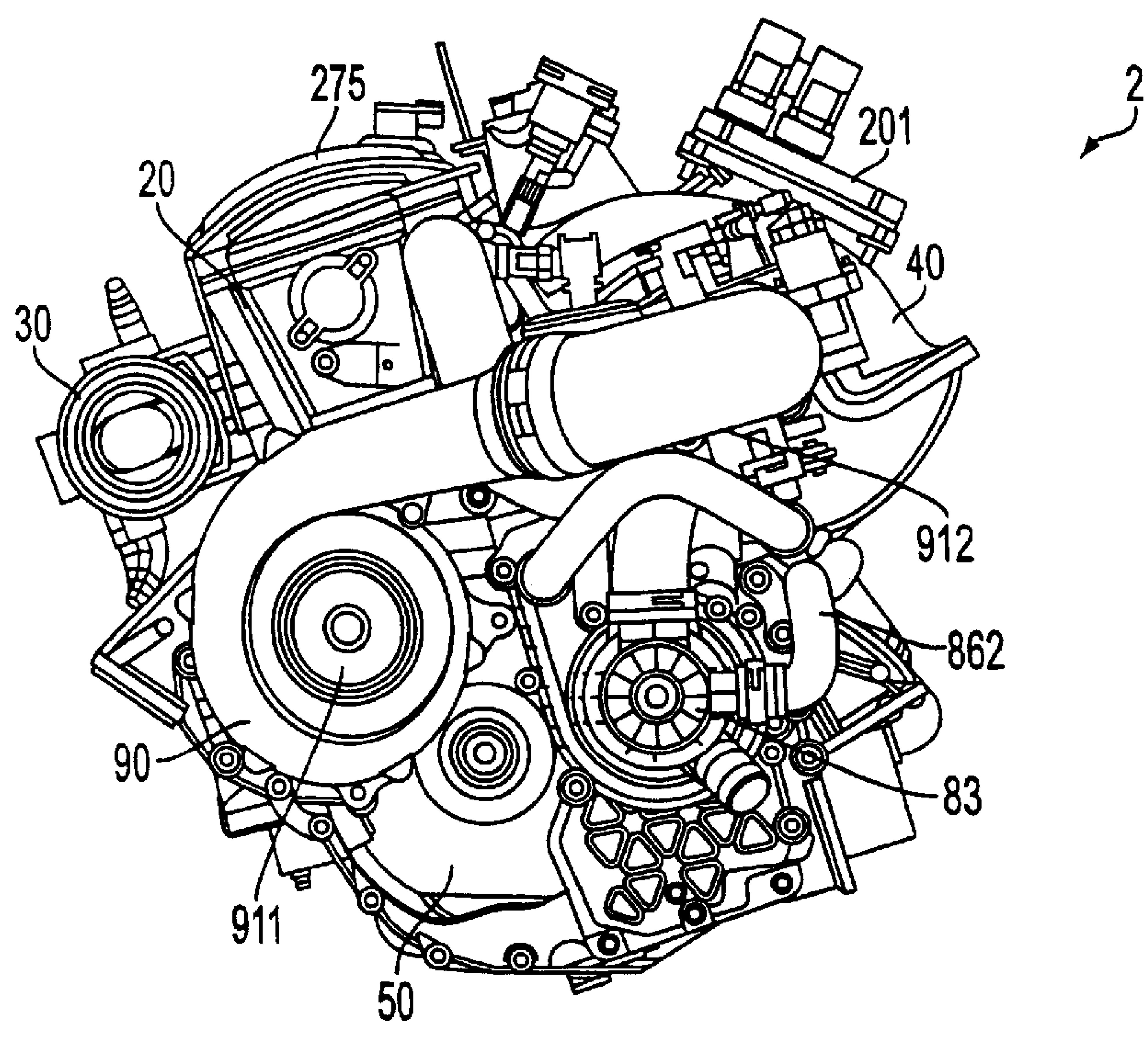

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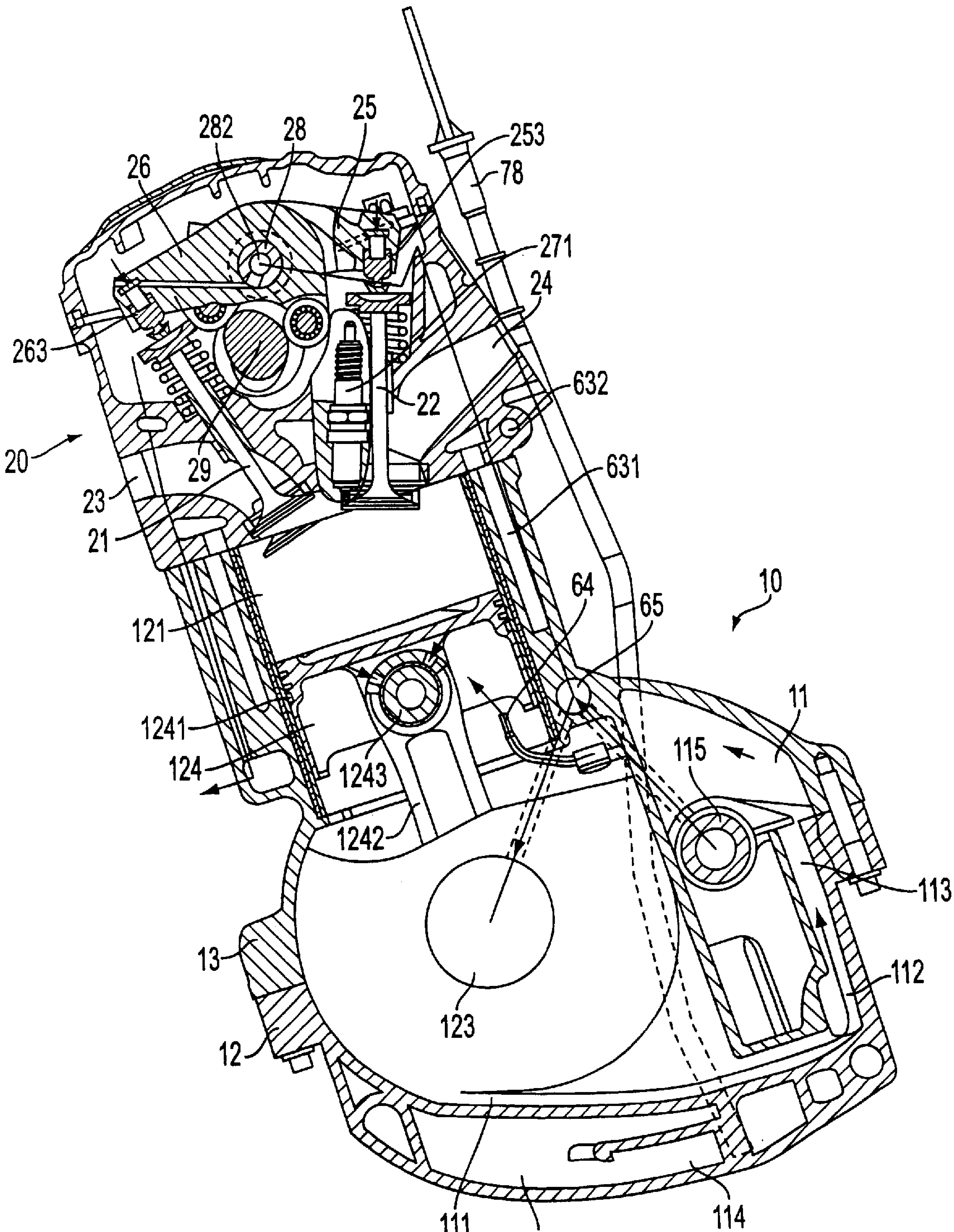
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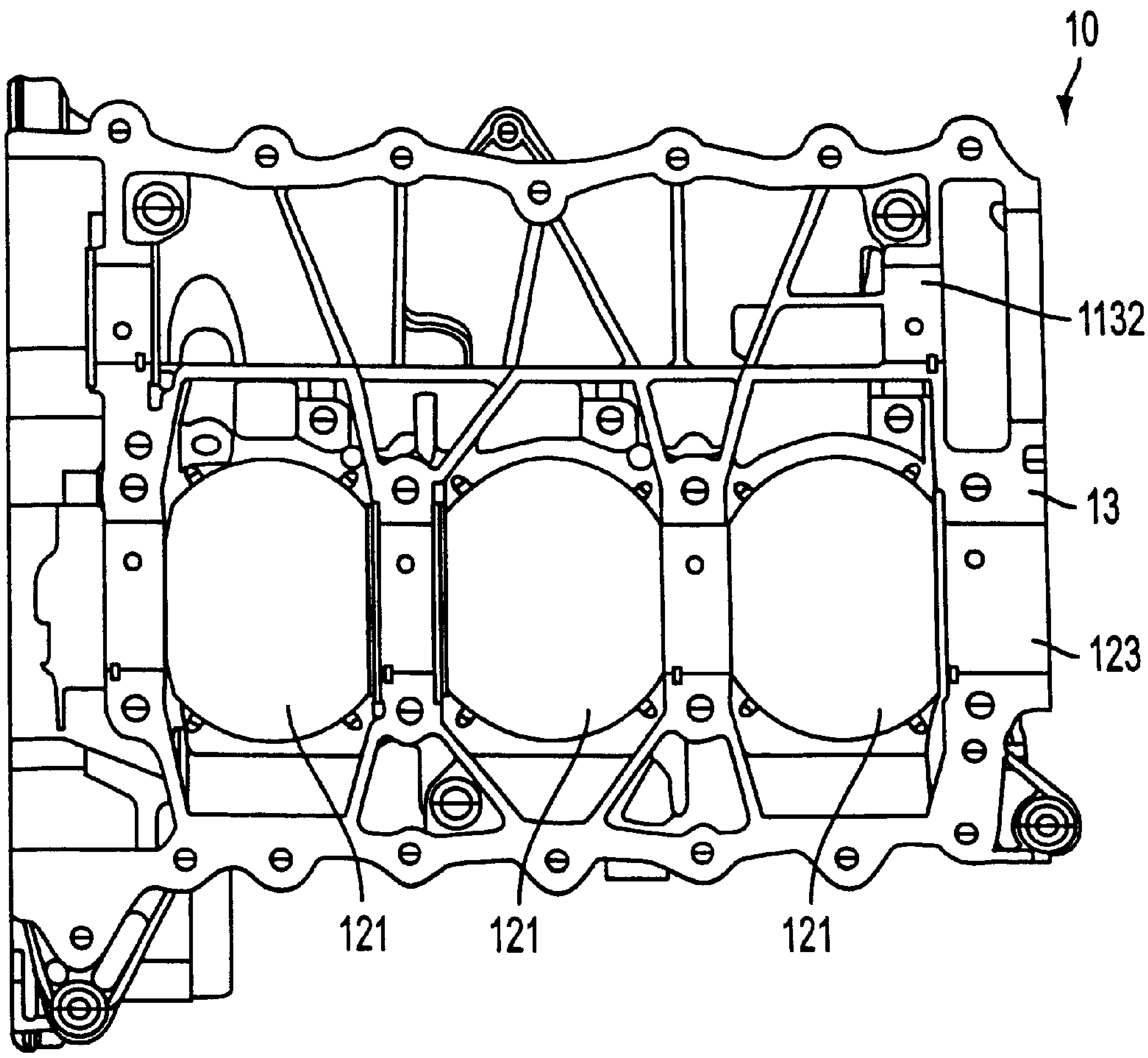
FIG. 8



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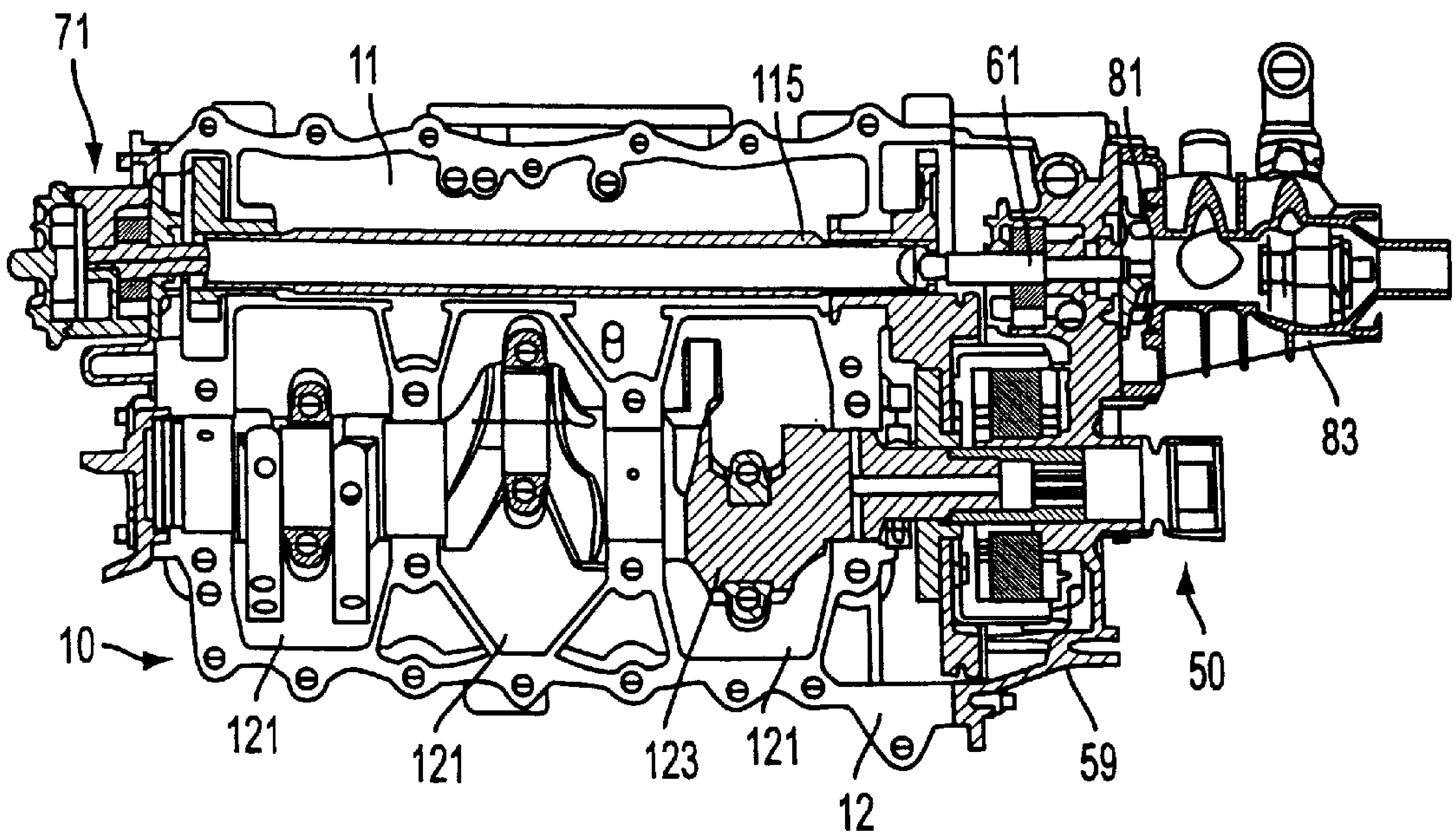

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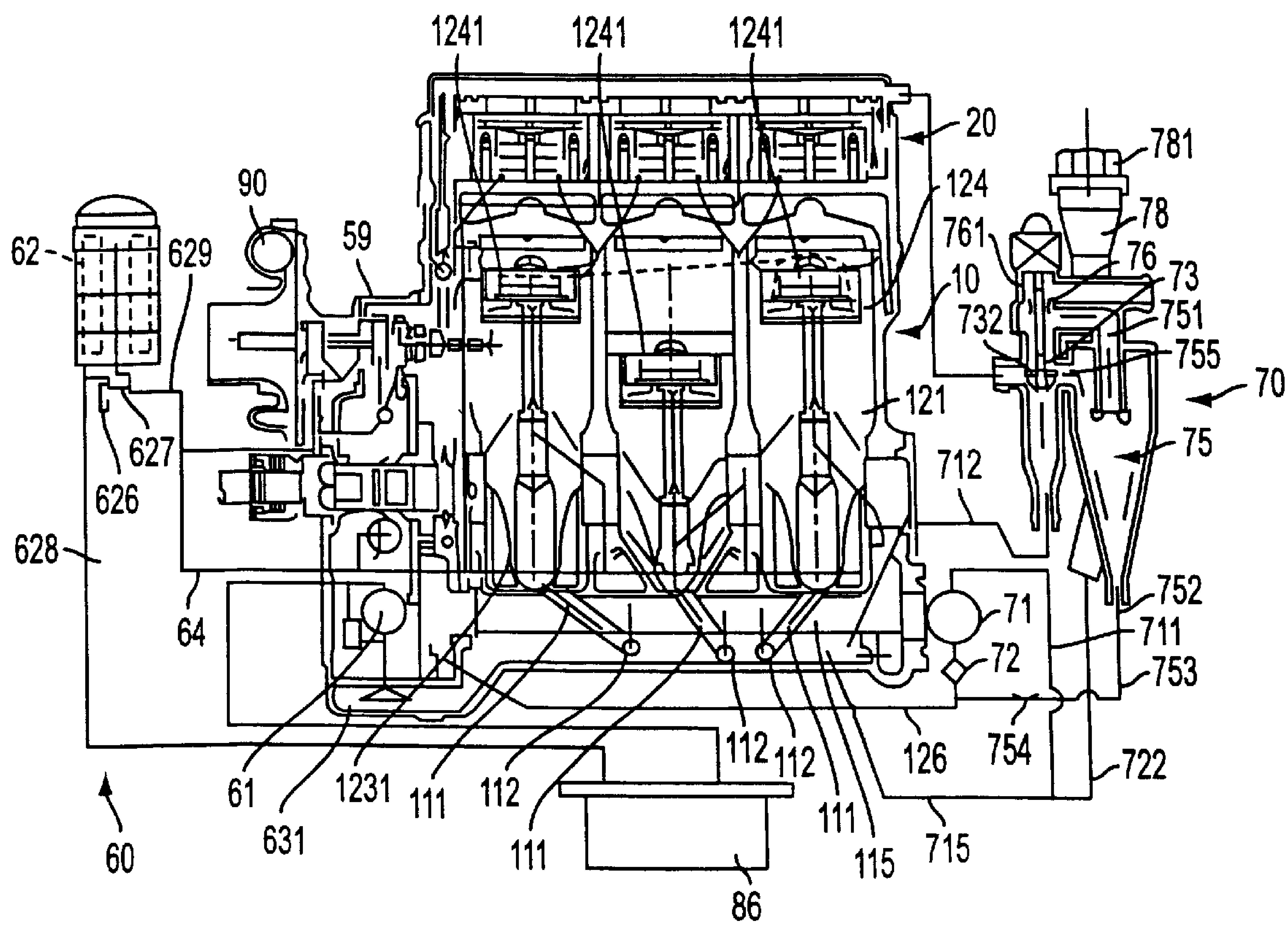
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FIG. 11

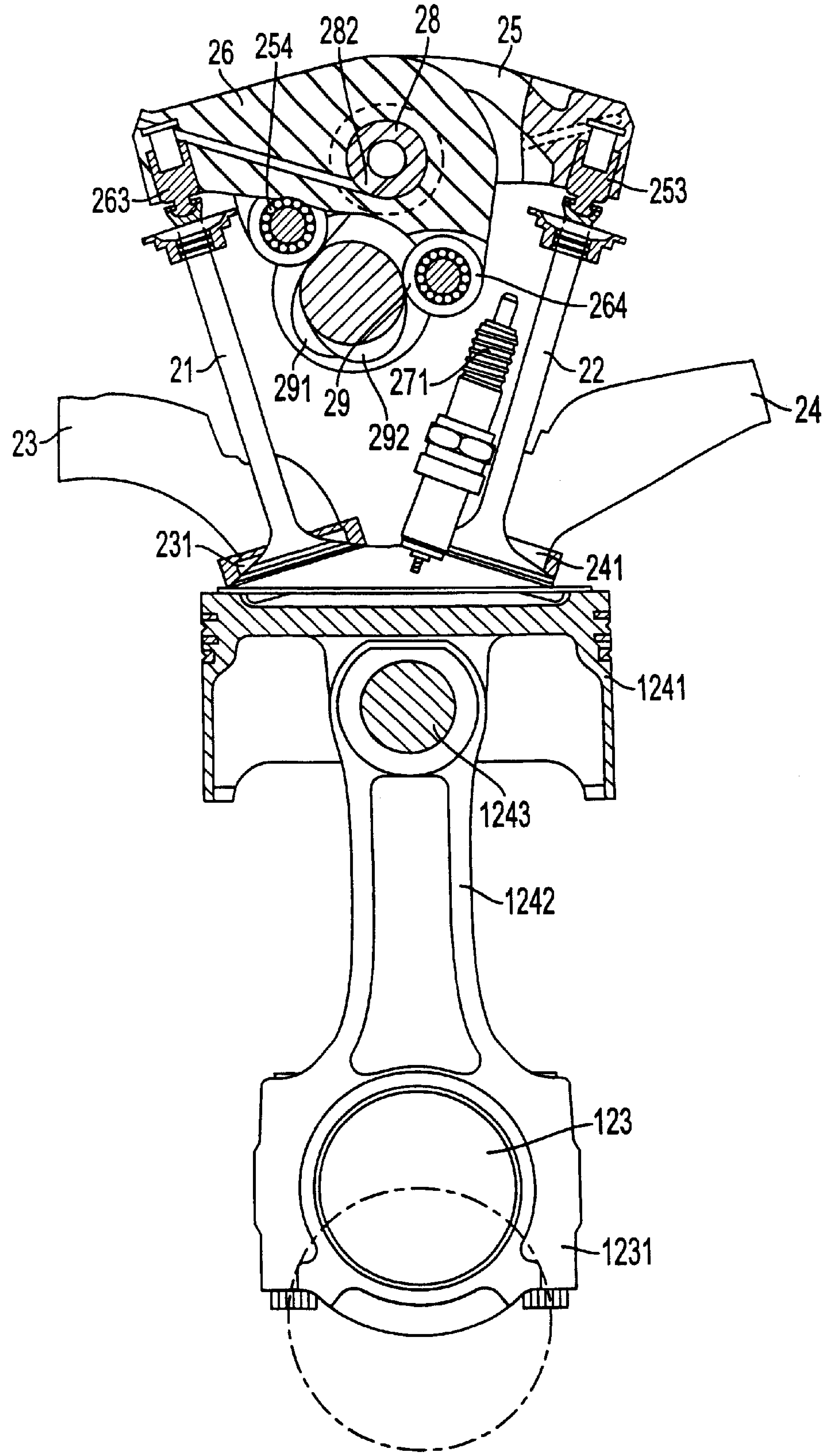
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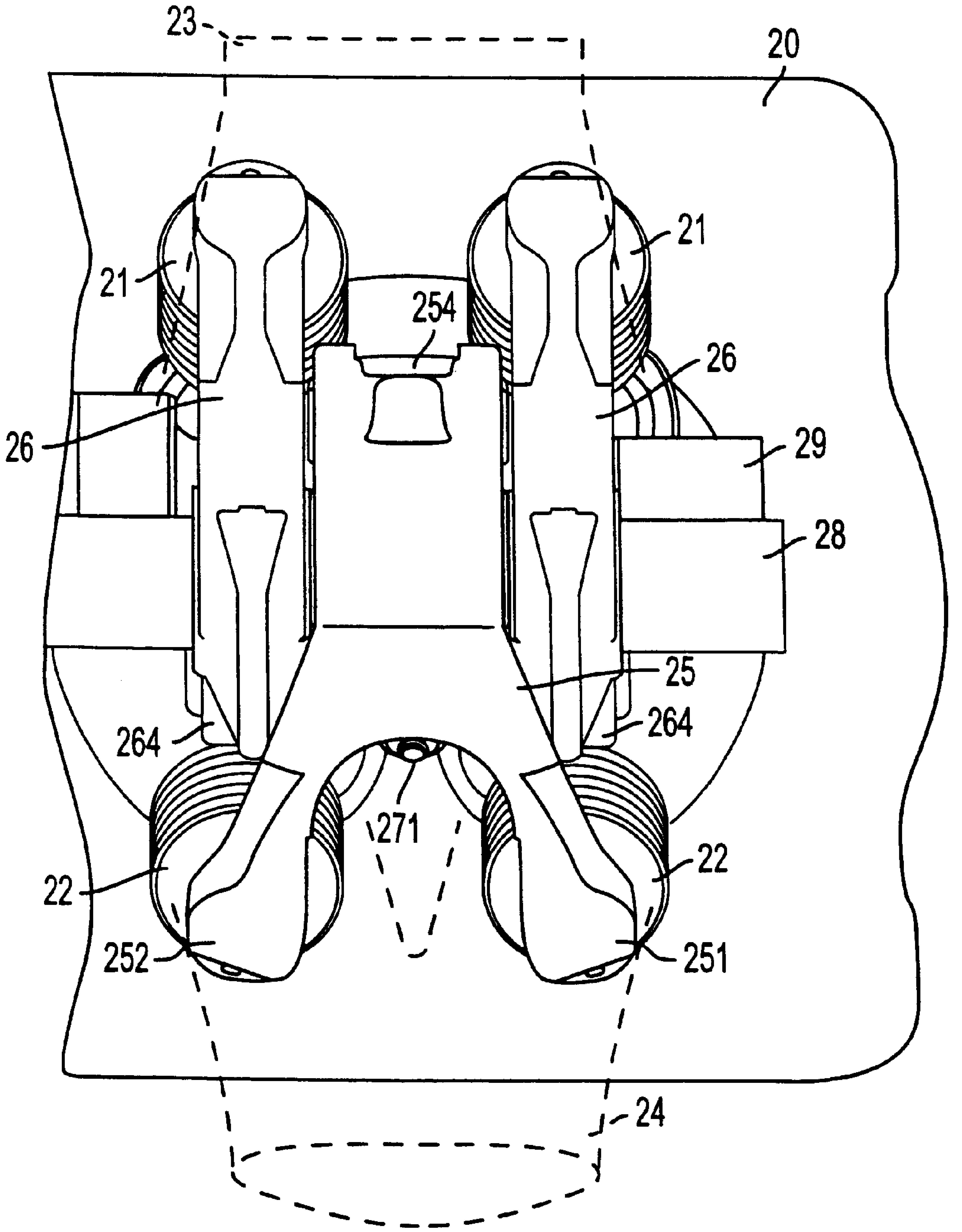
FIG. 12



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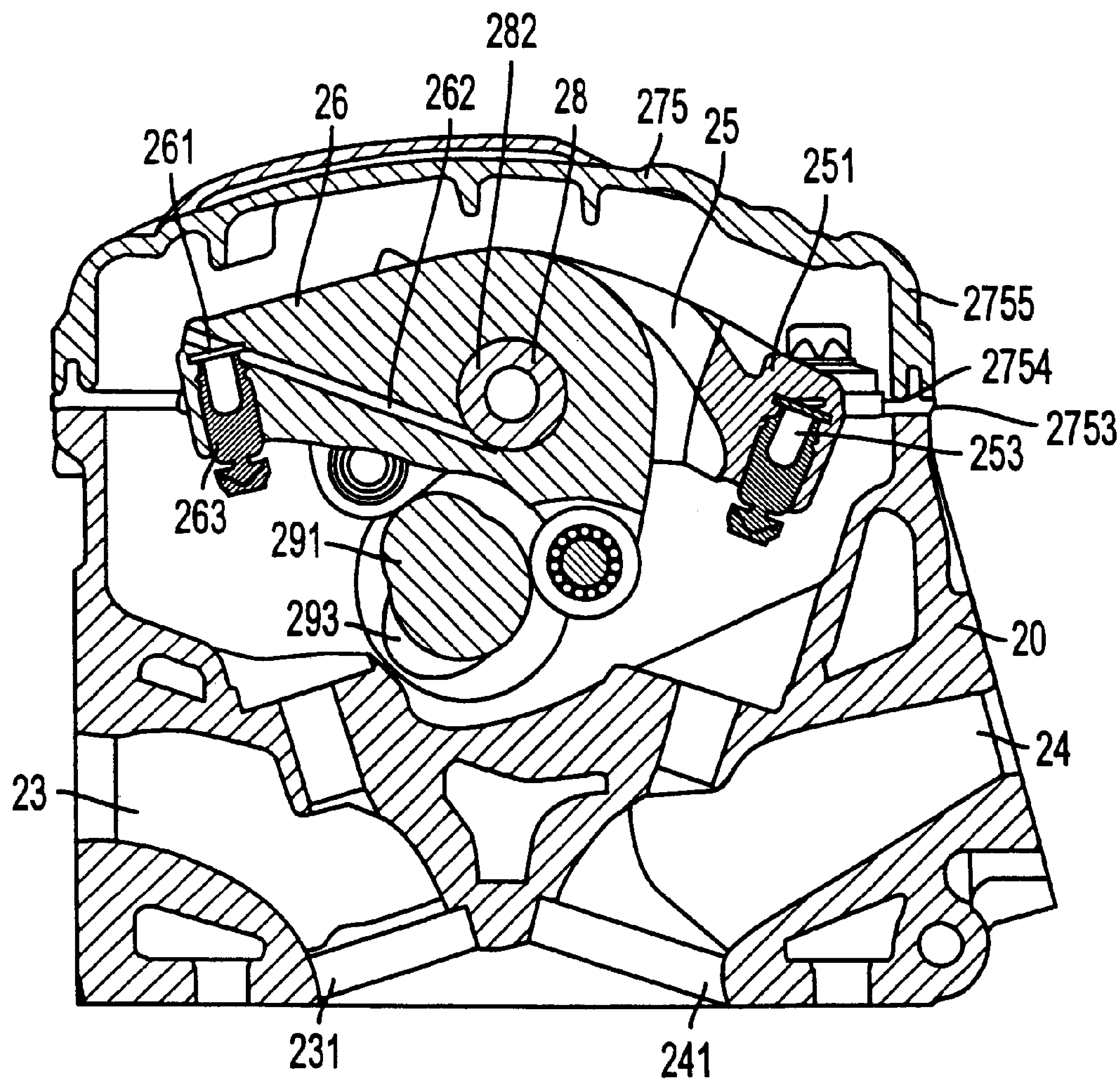
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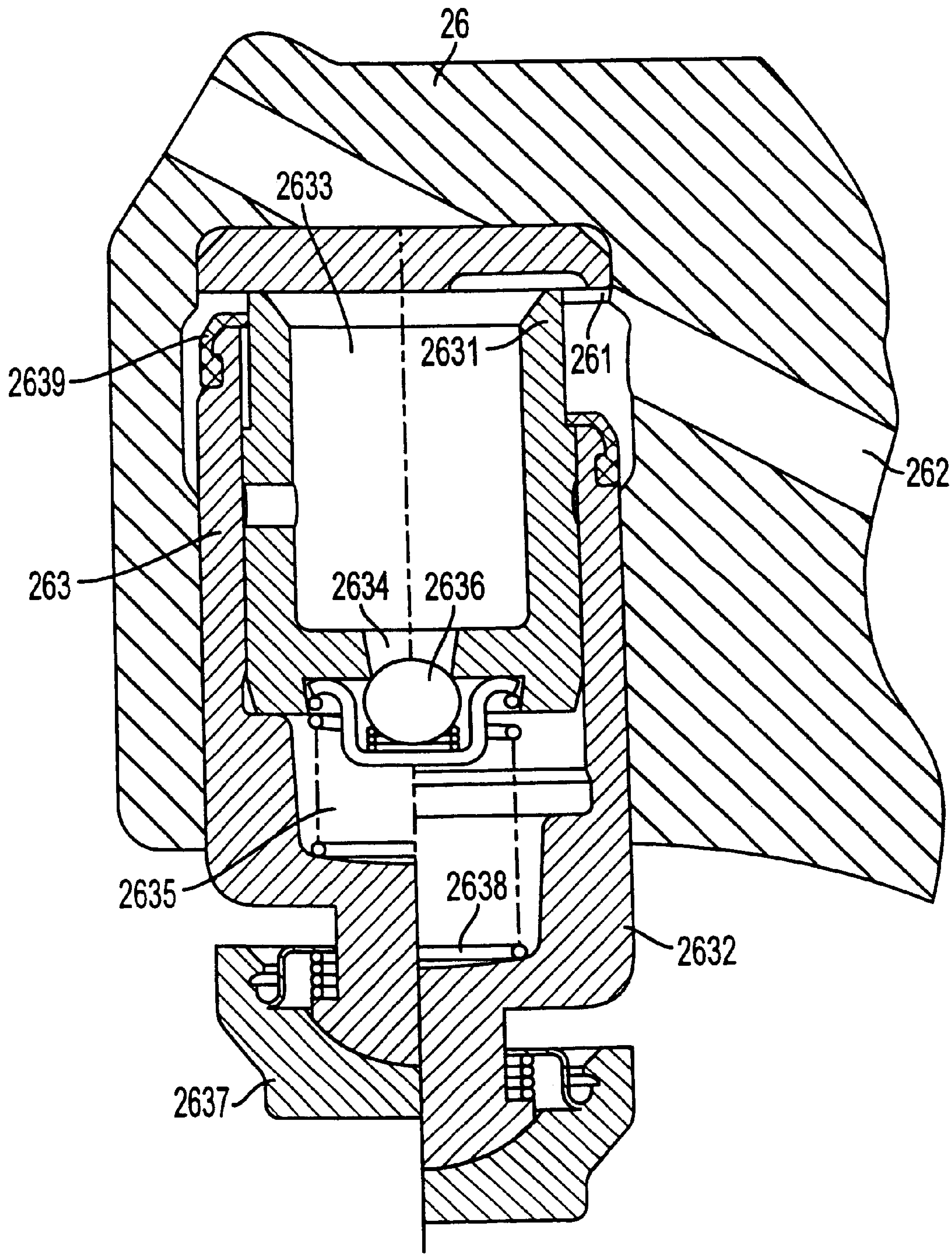
FIG. 14



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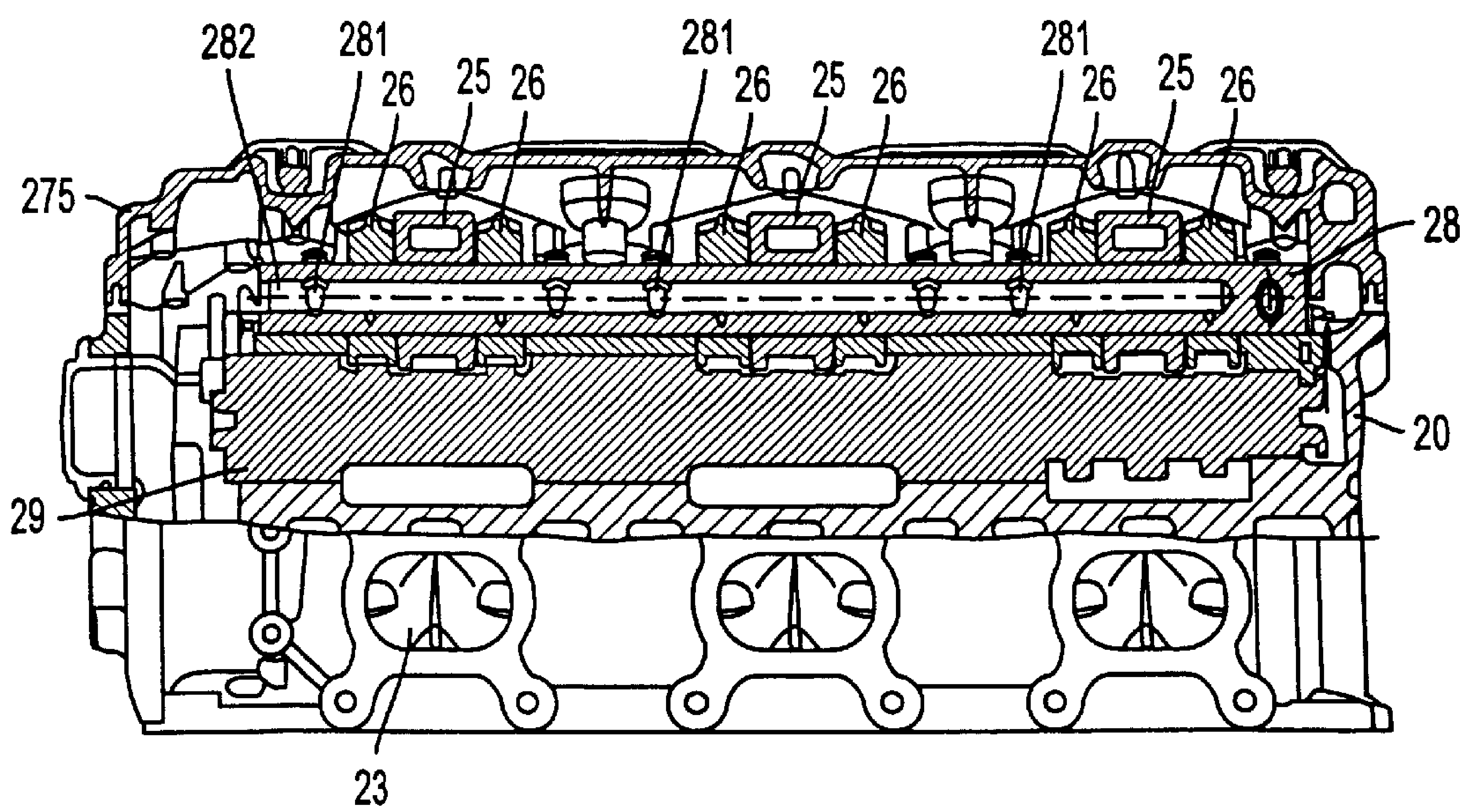

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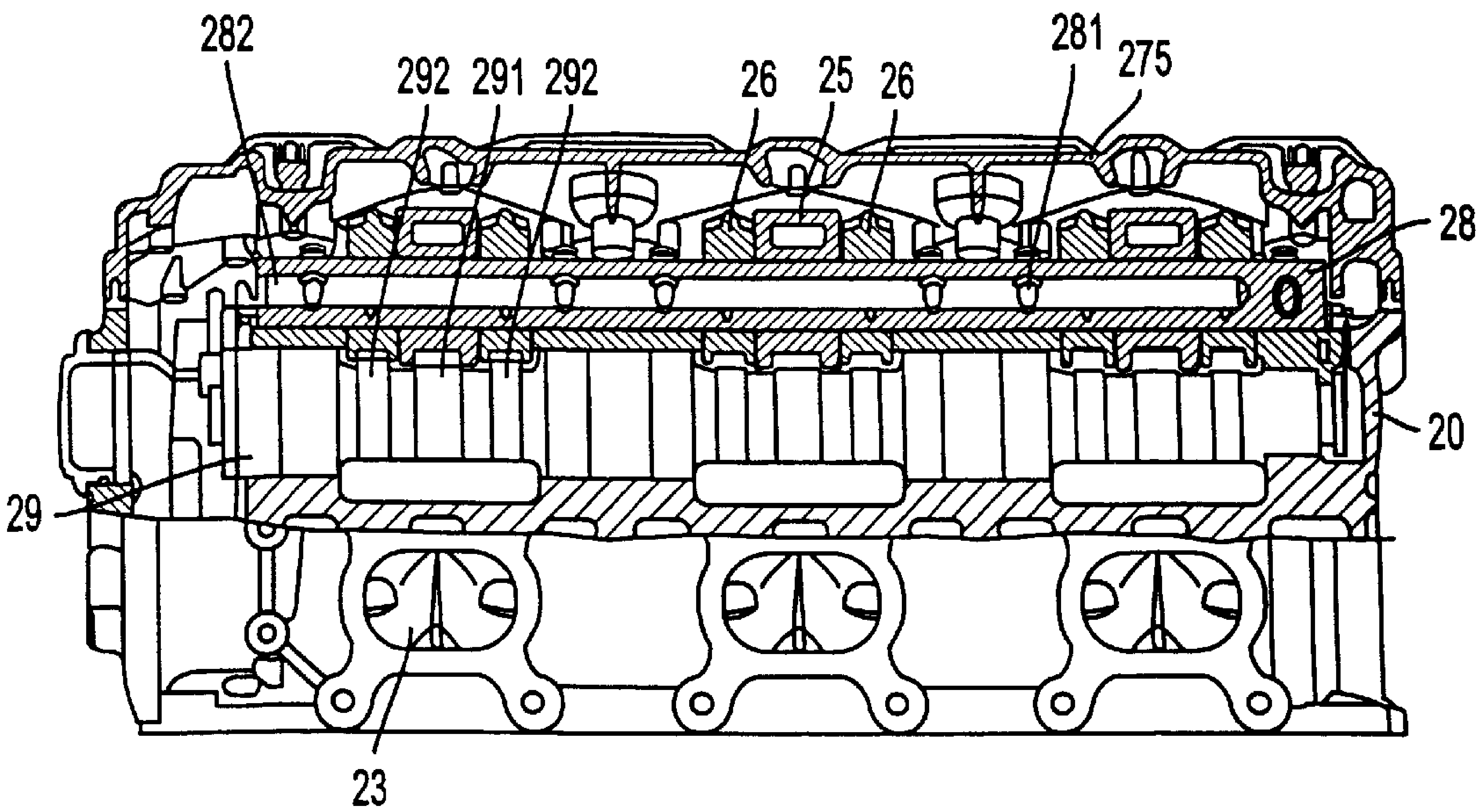

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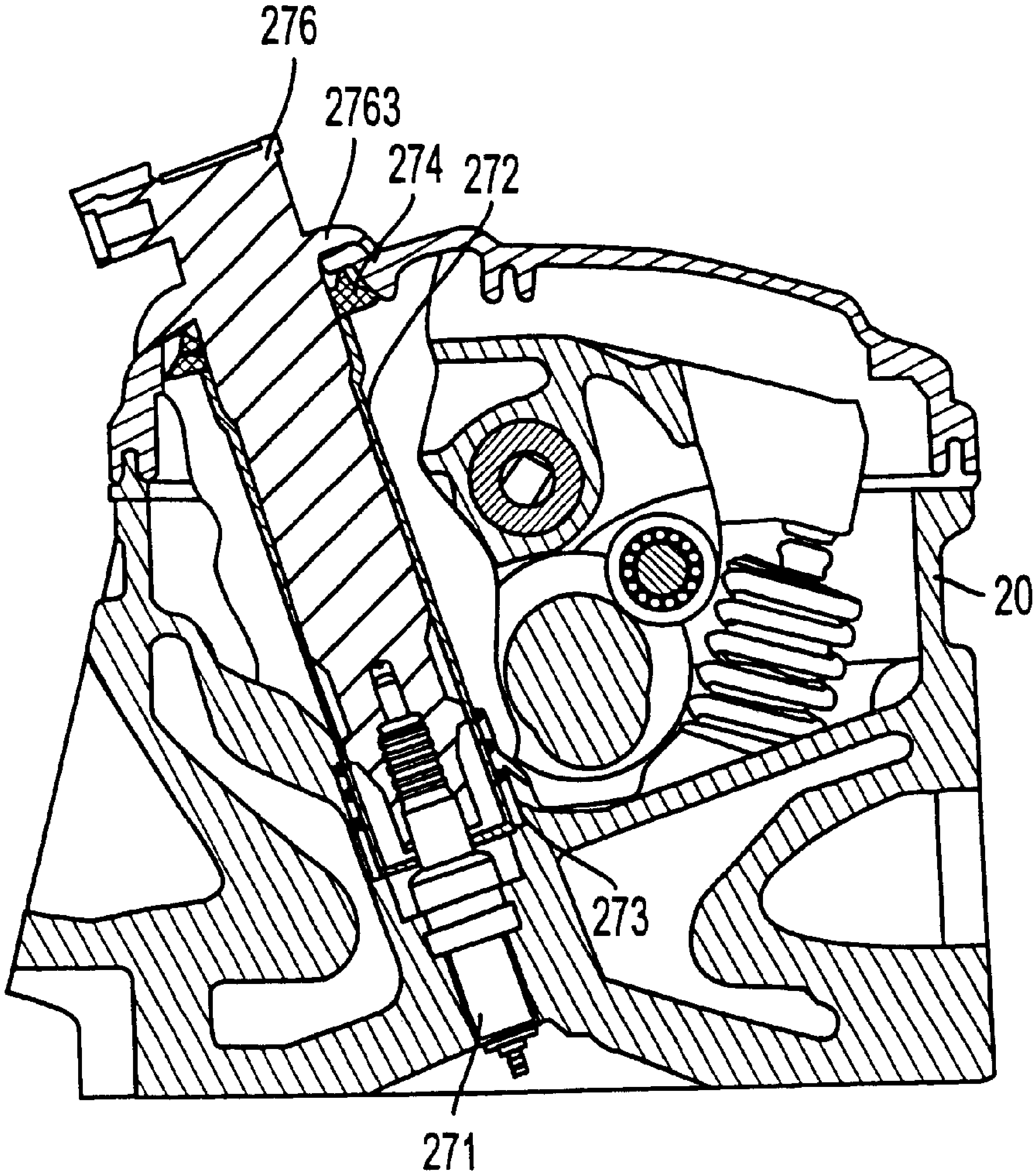

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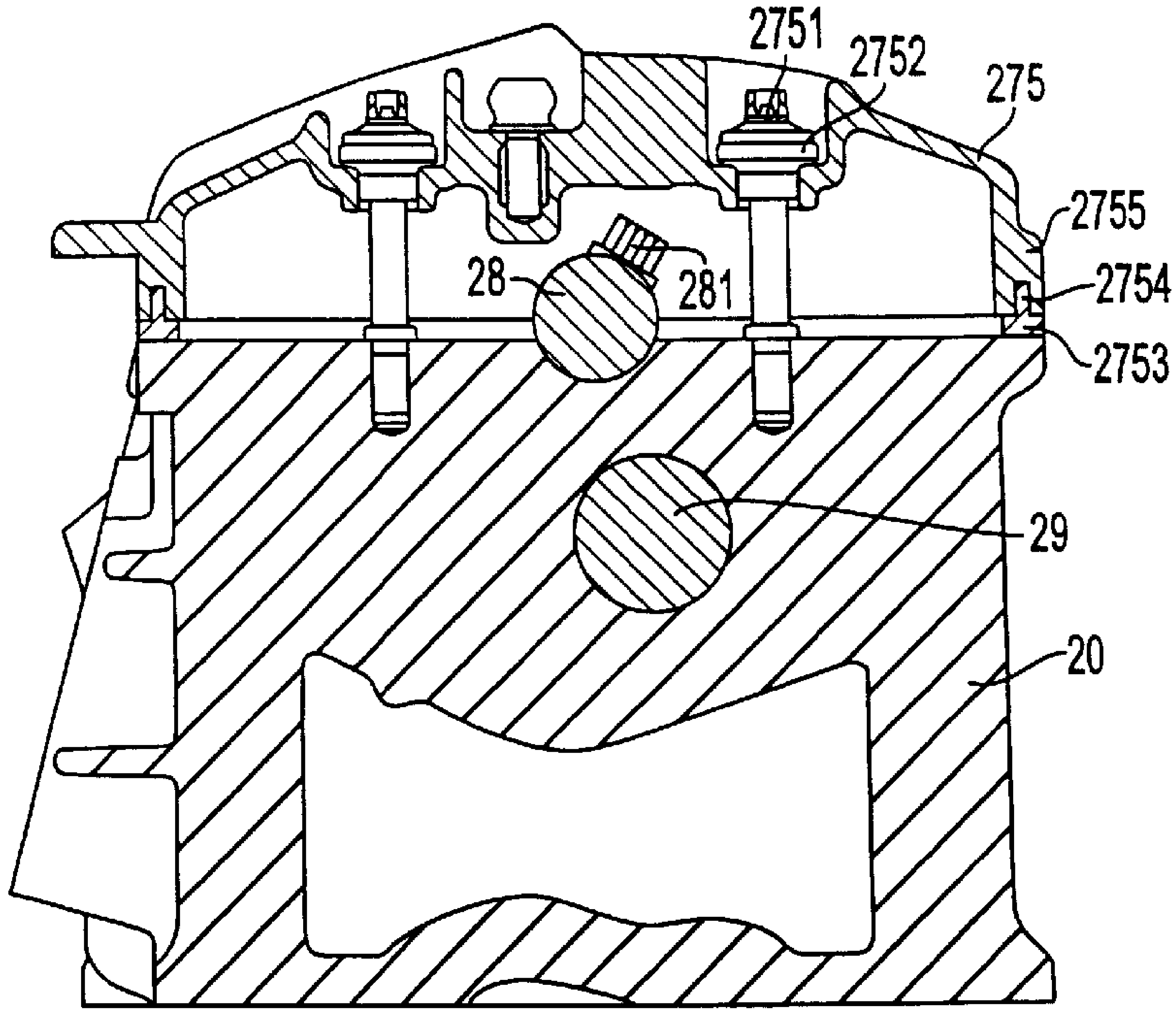
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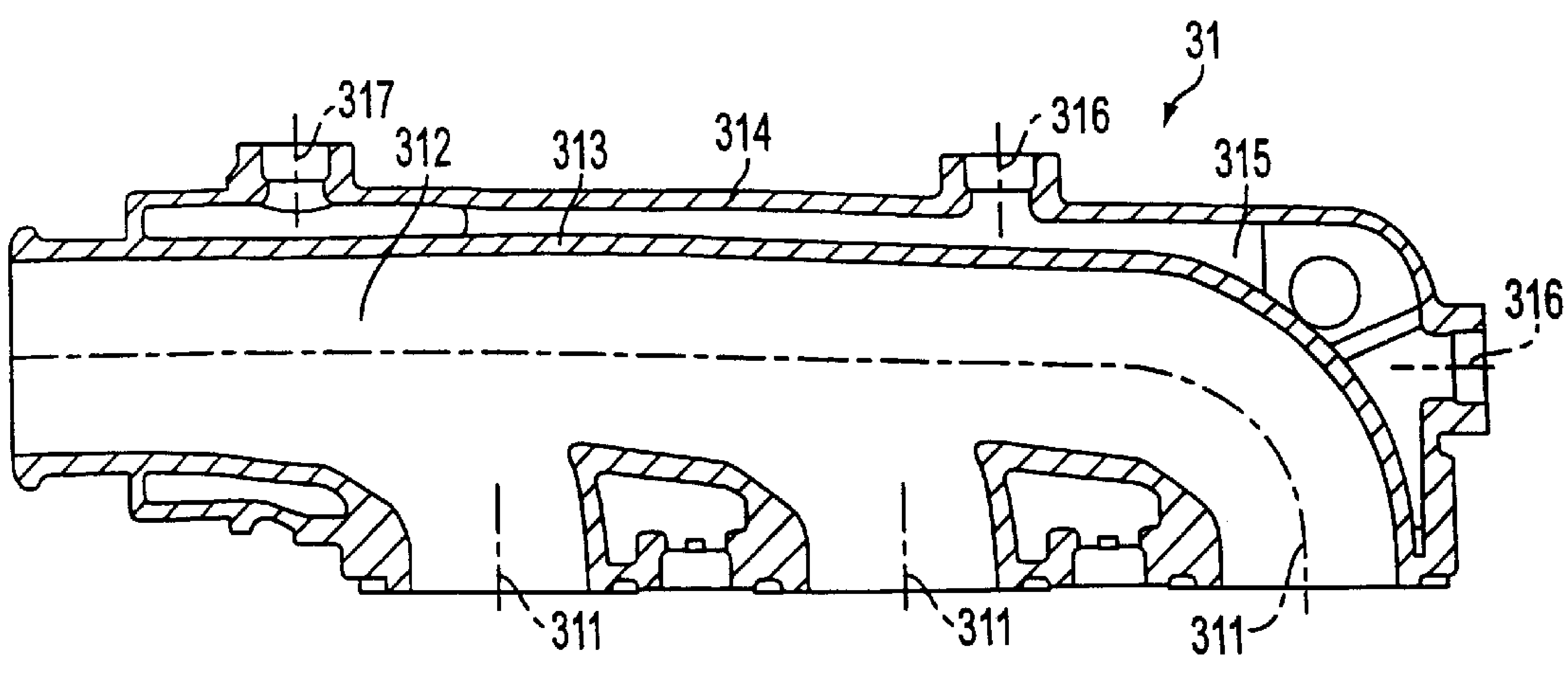
FIG. 19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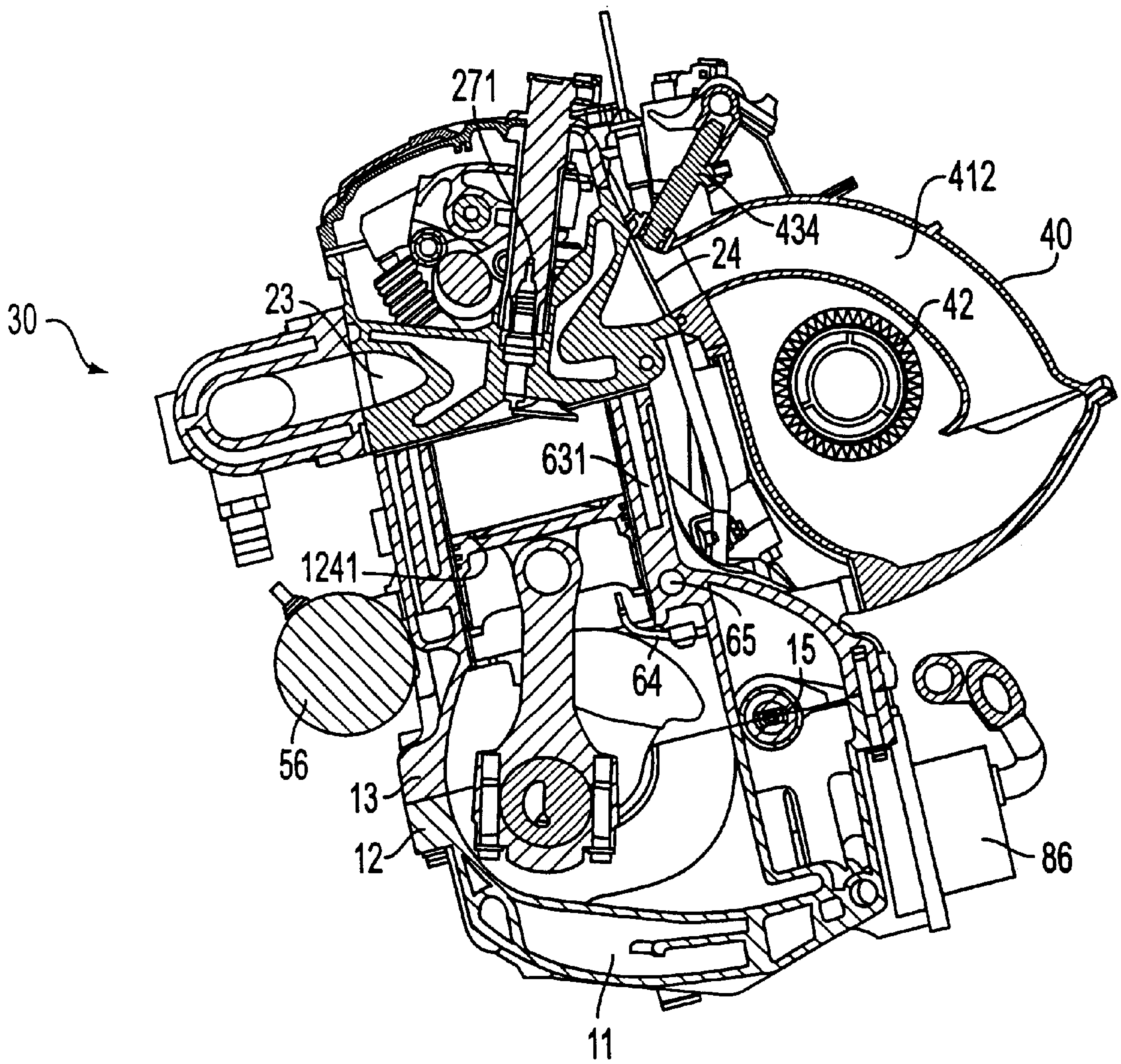

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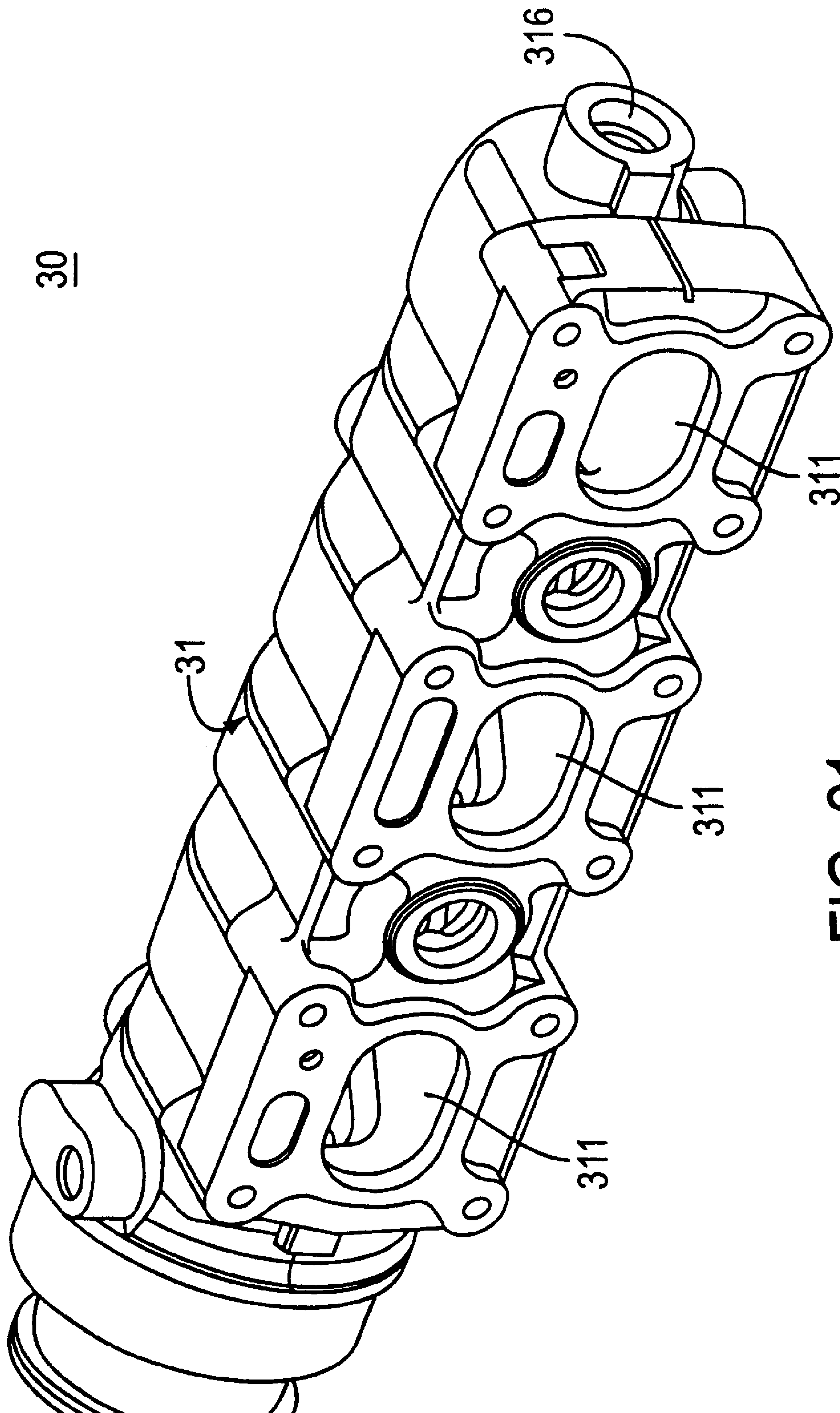

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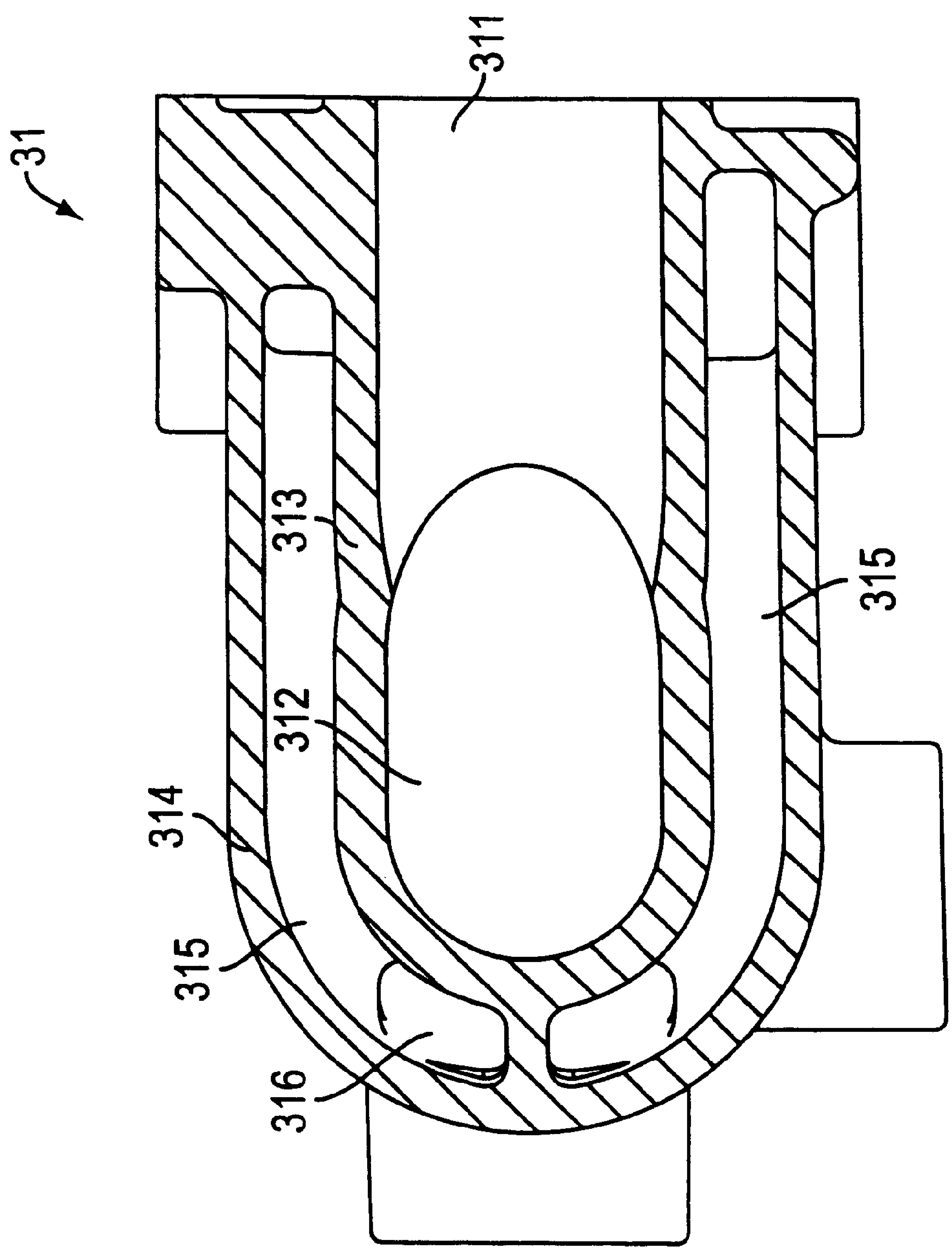

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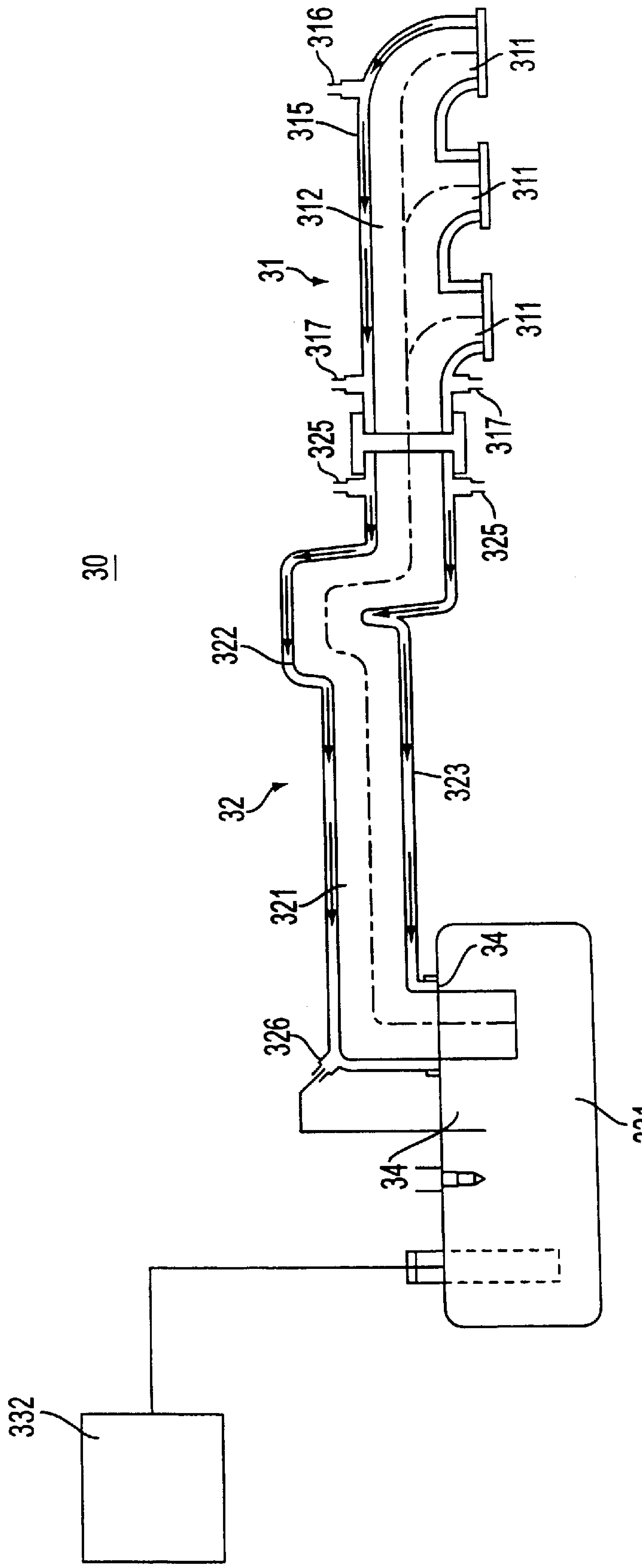
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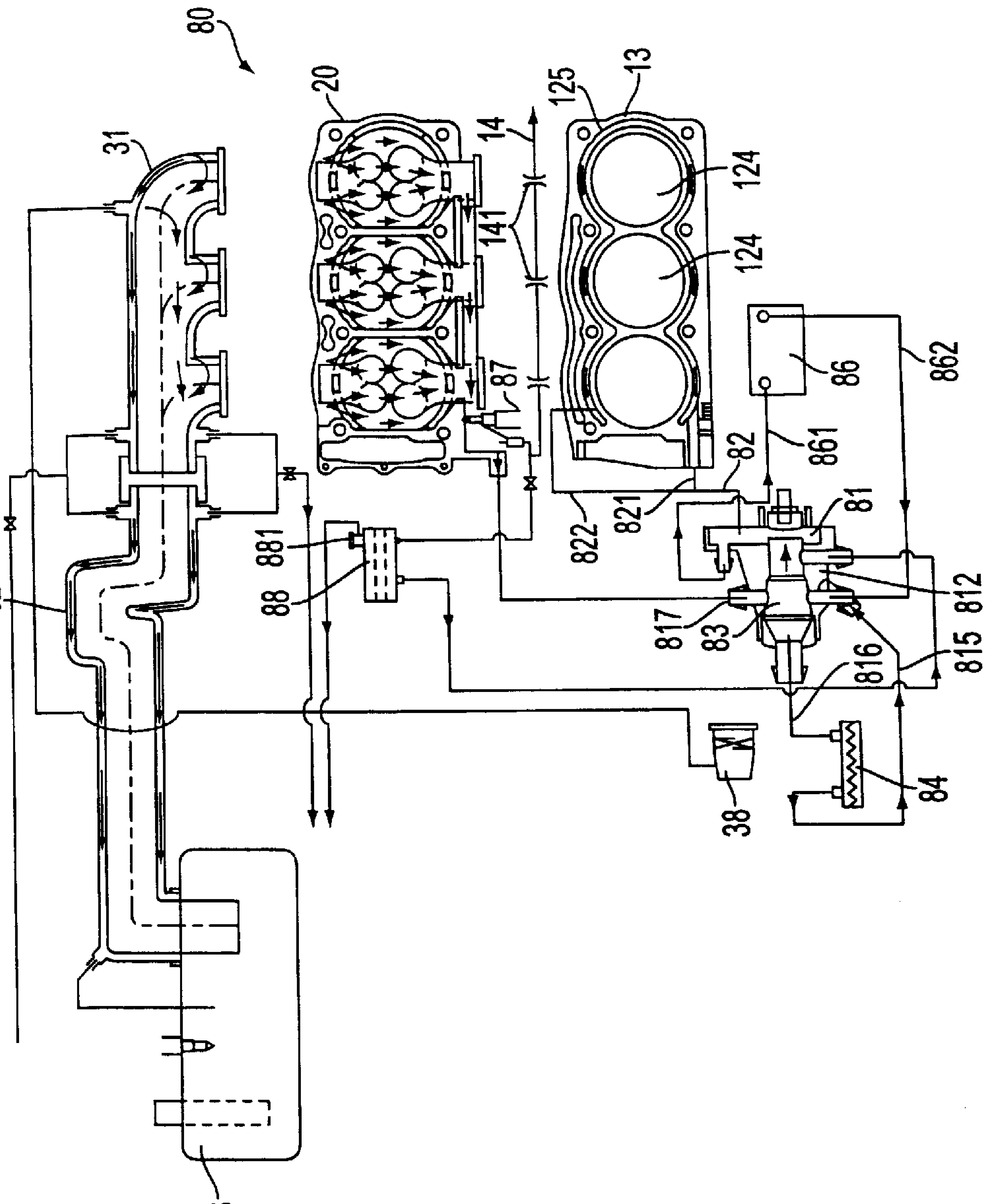
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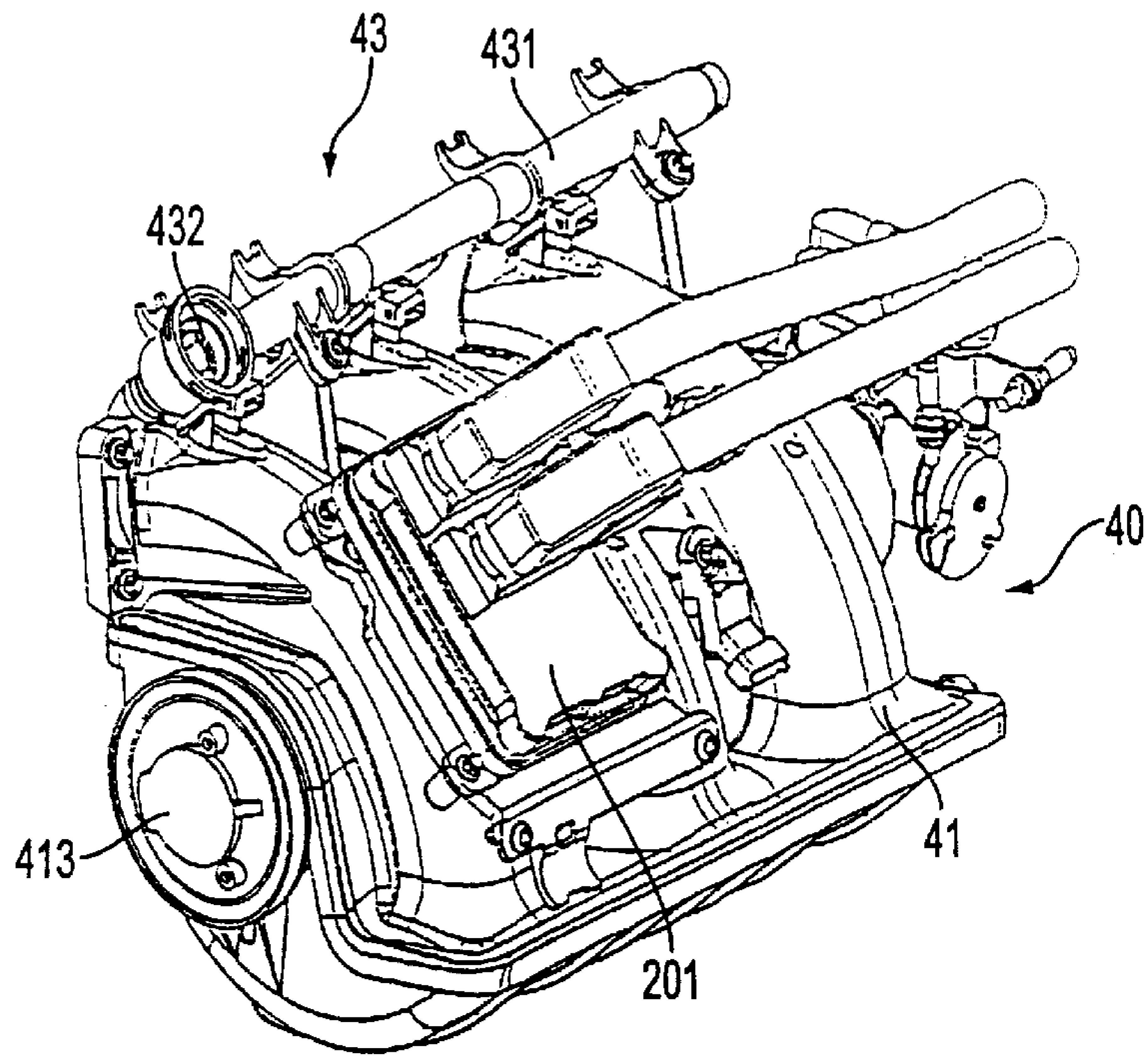
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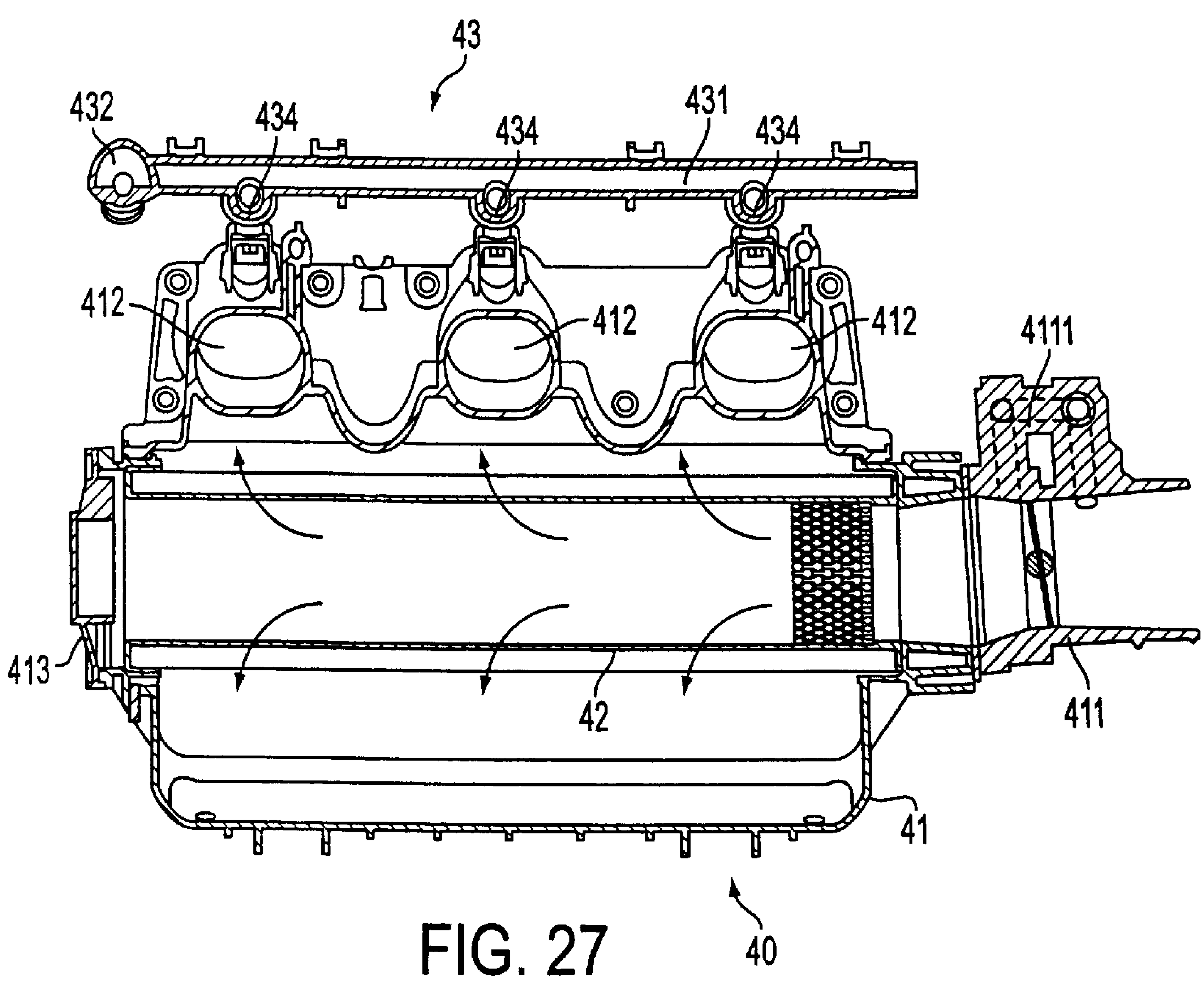
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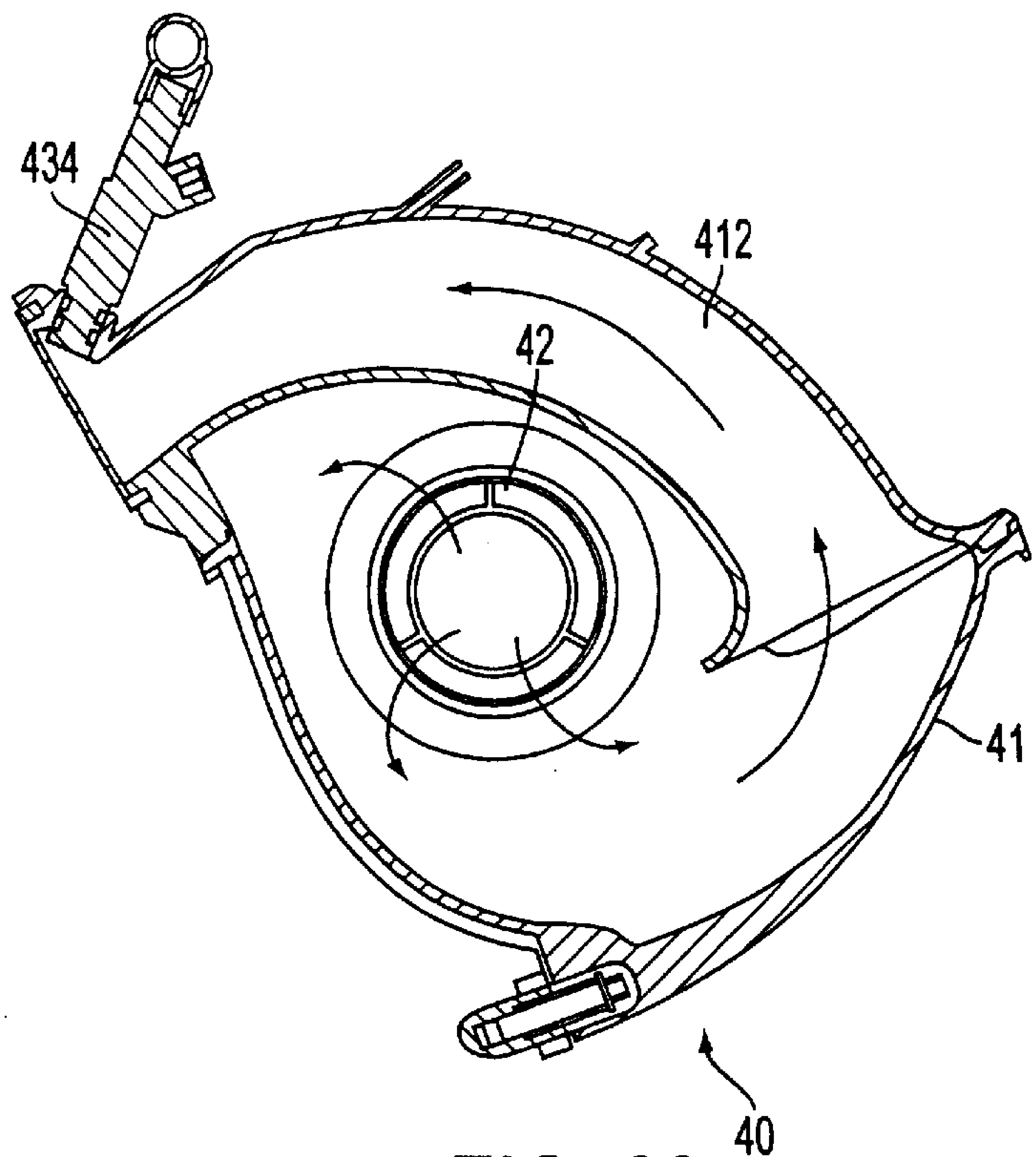

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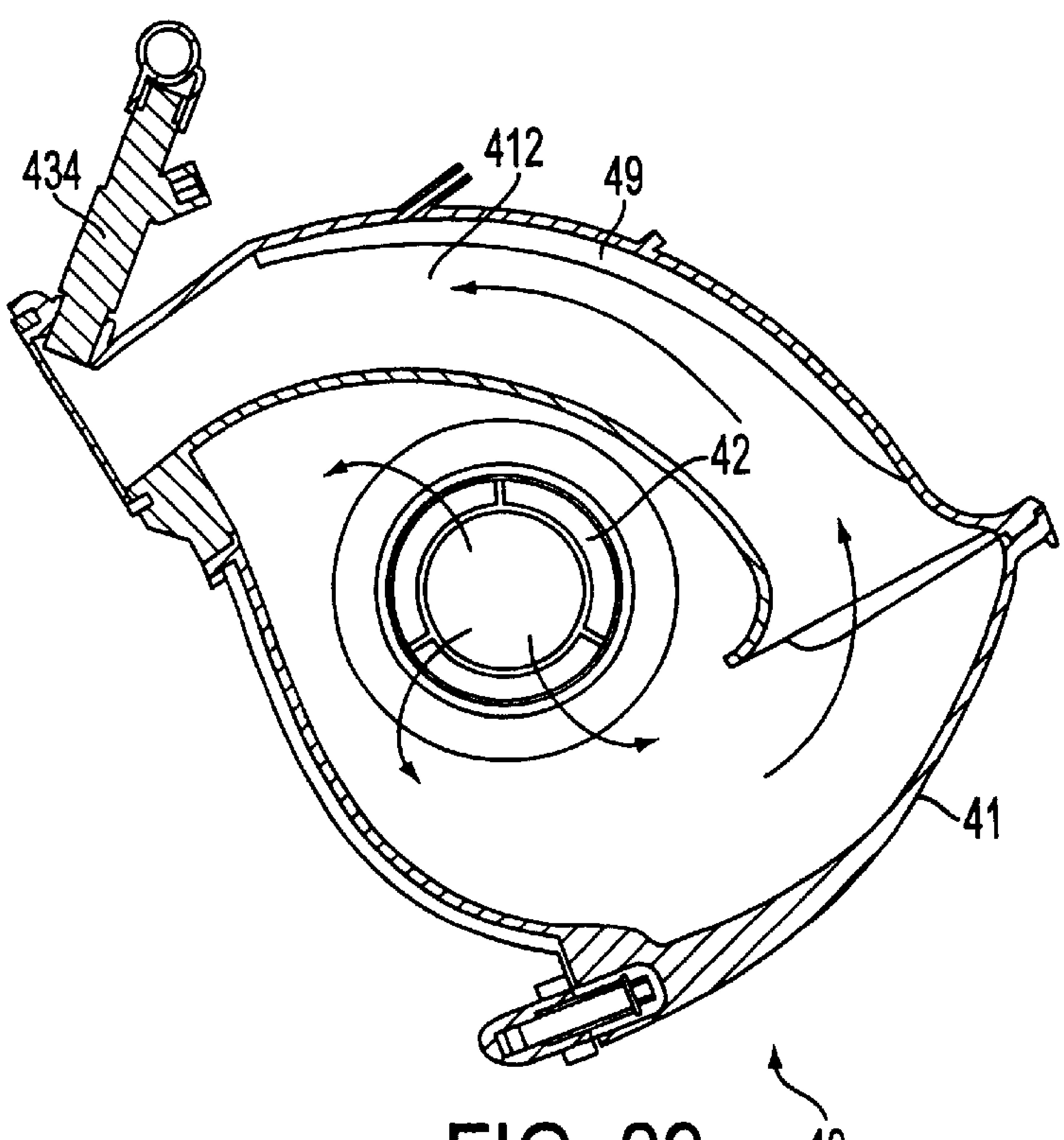

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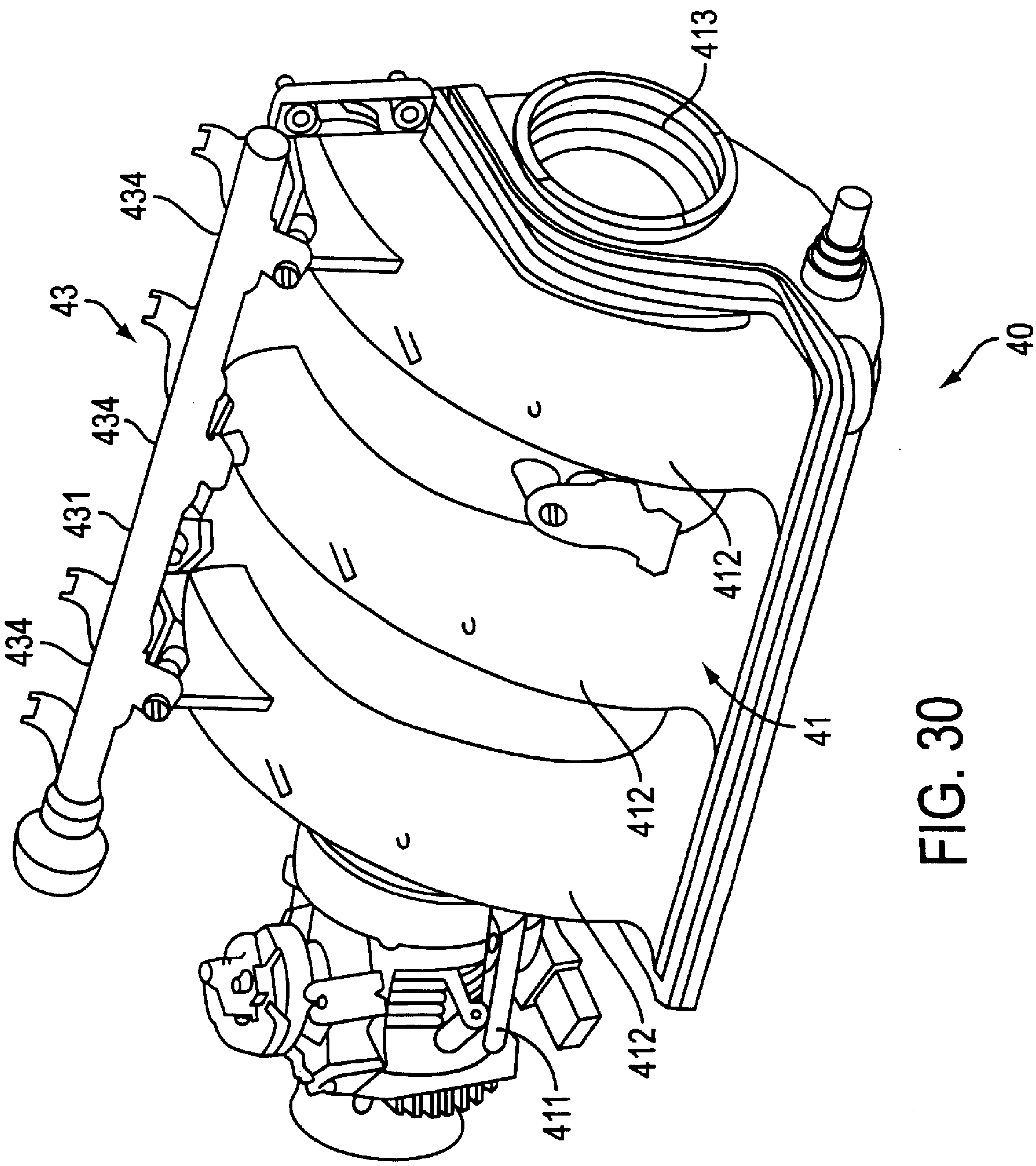

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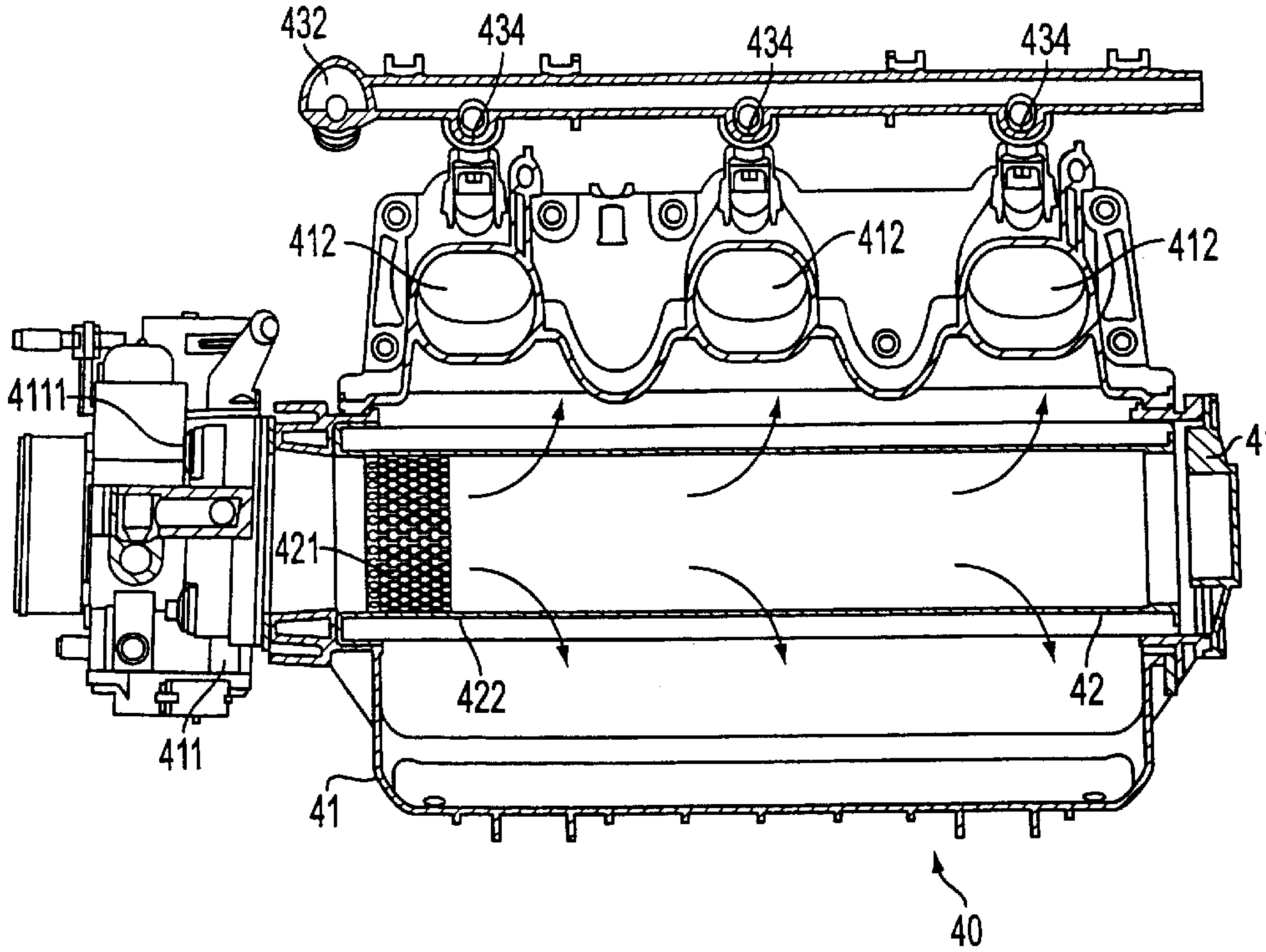
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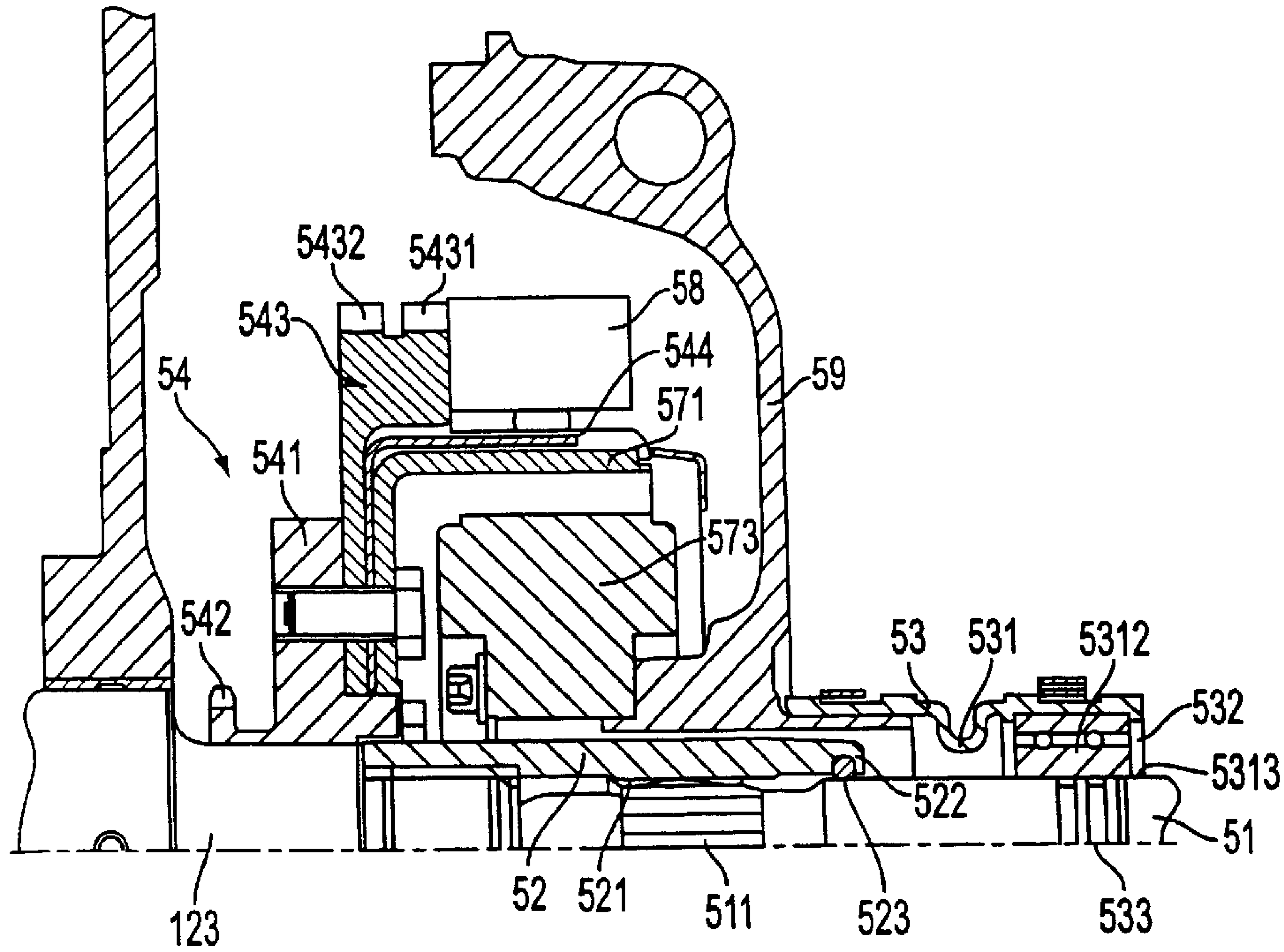
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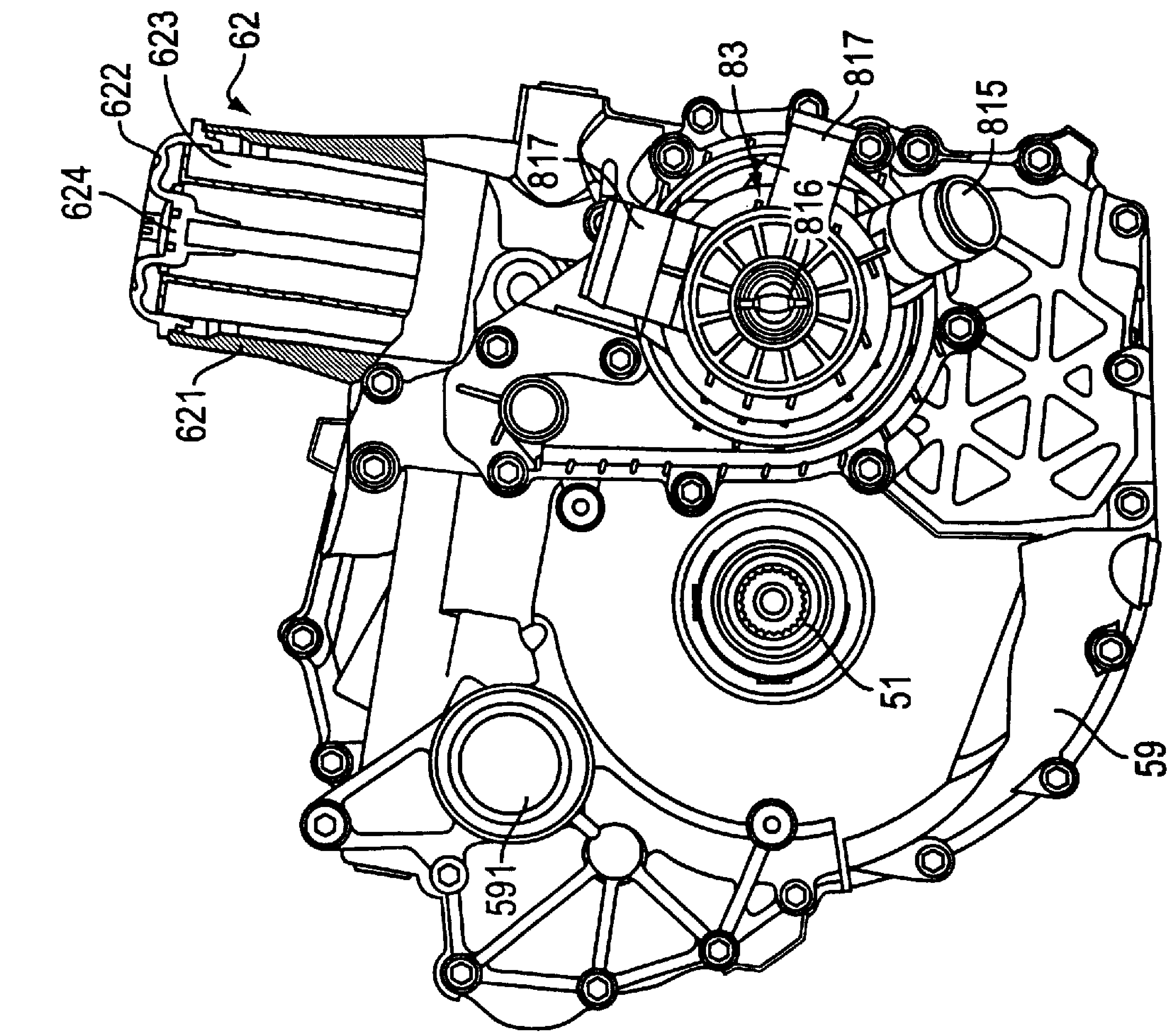

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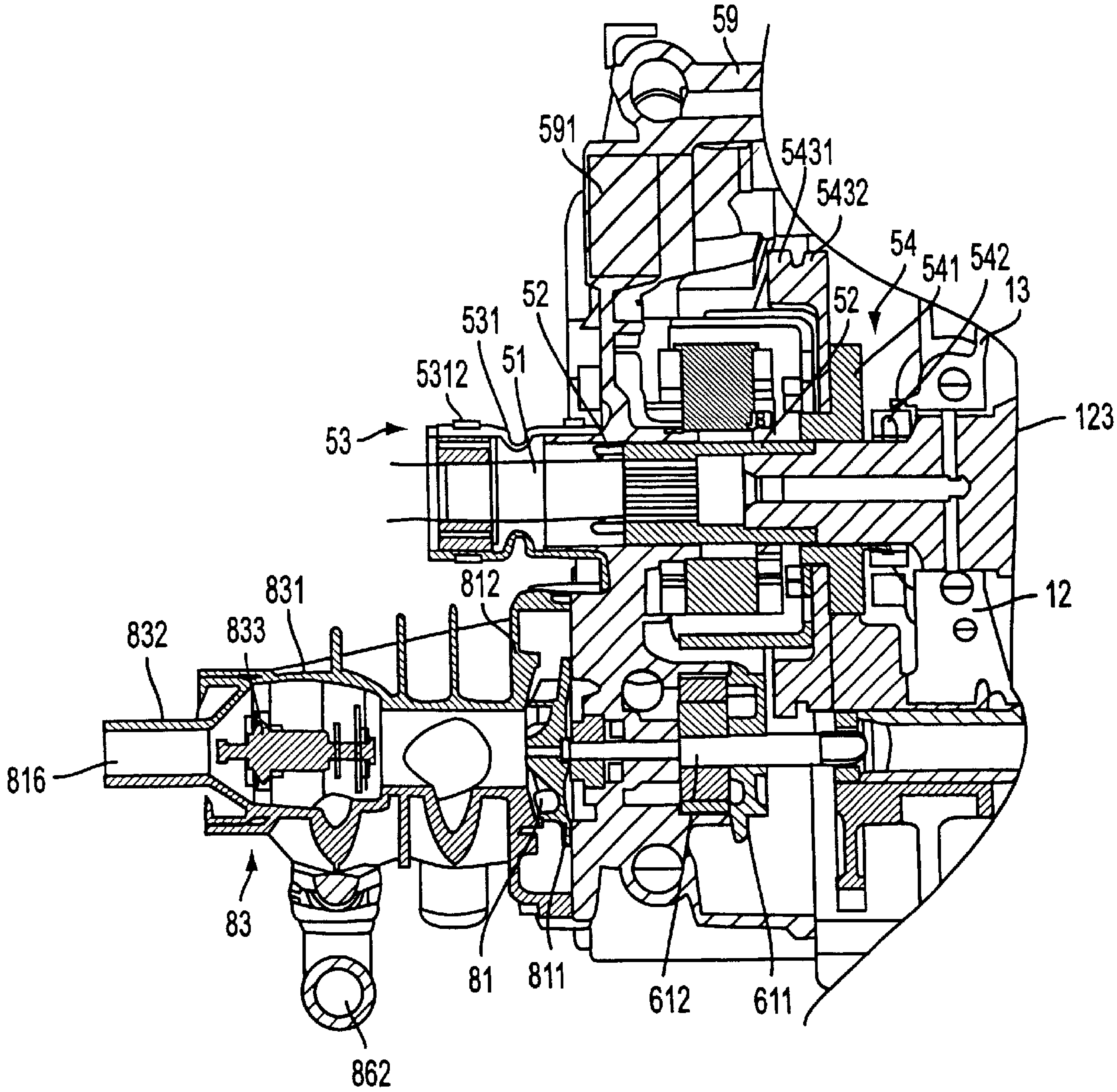
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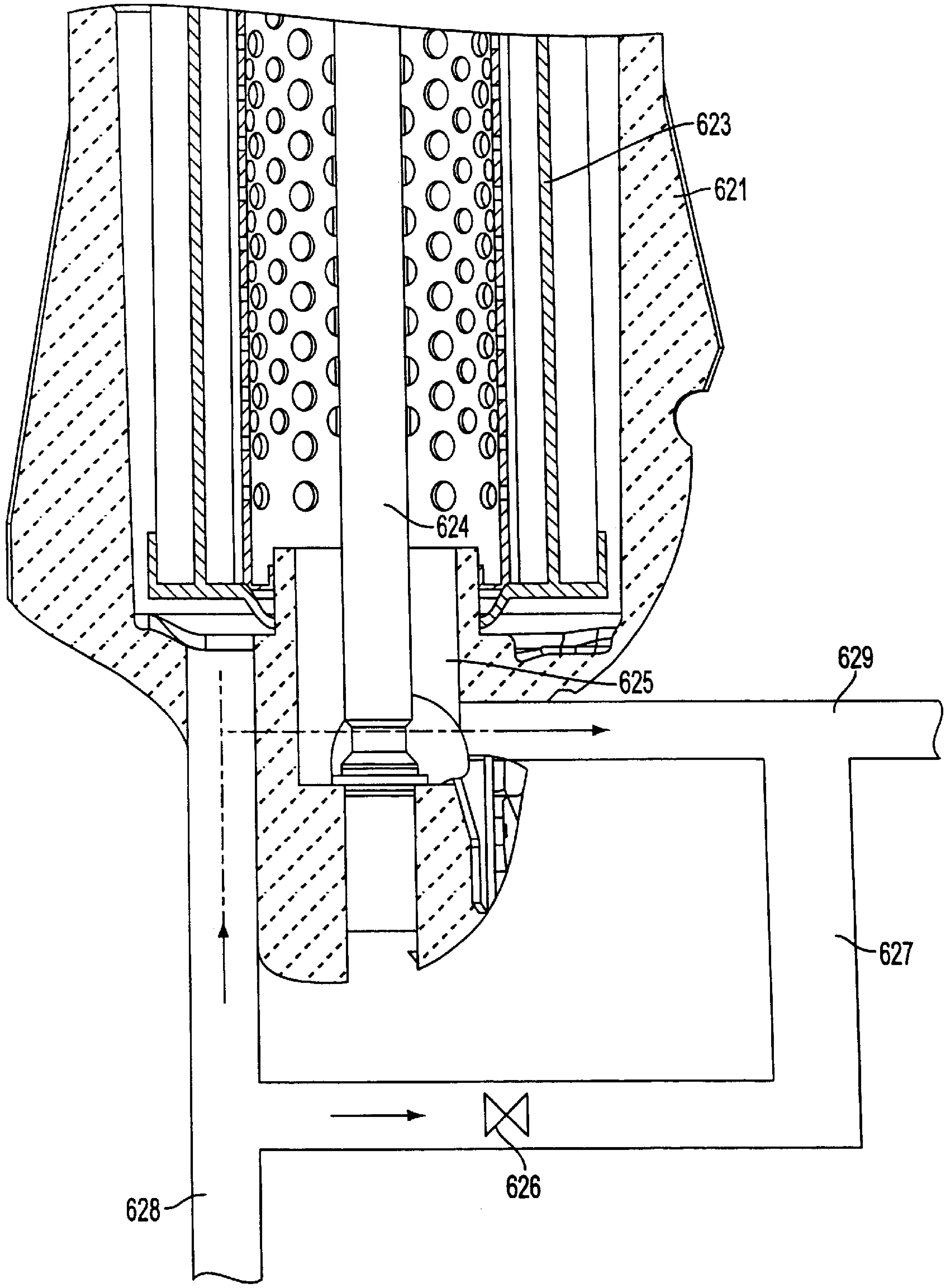
FIG. 33



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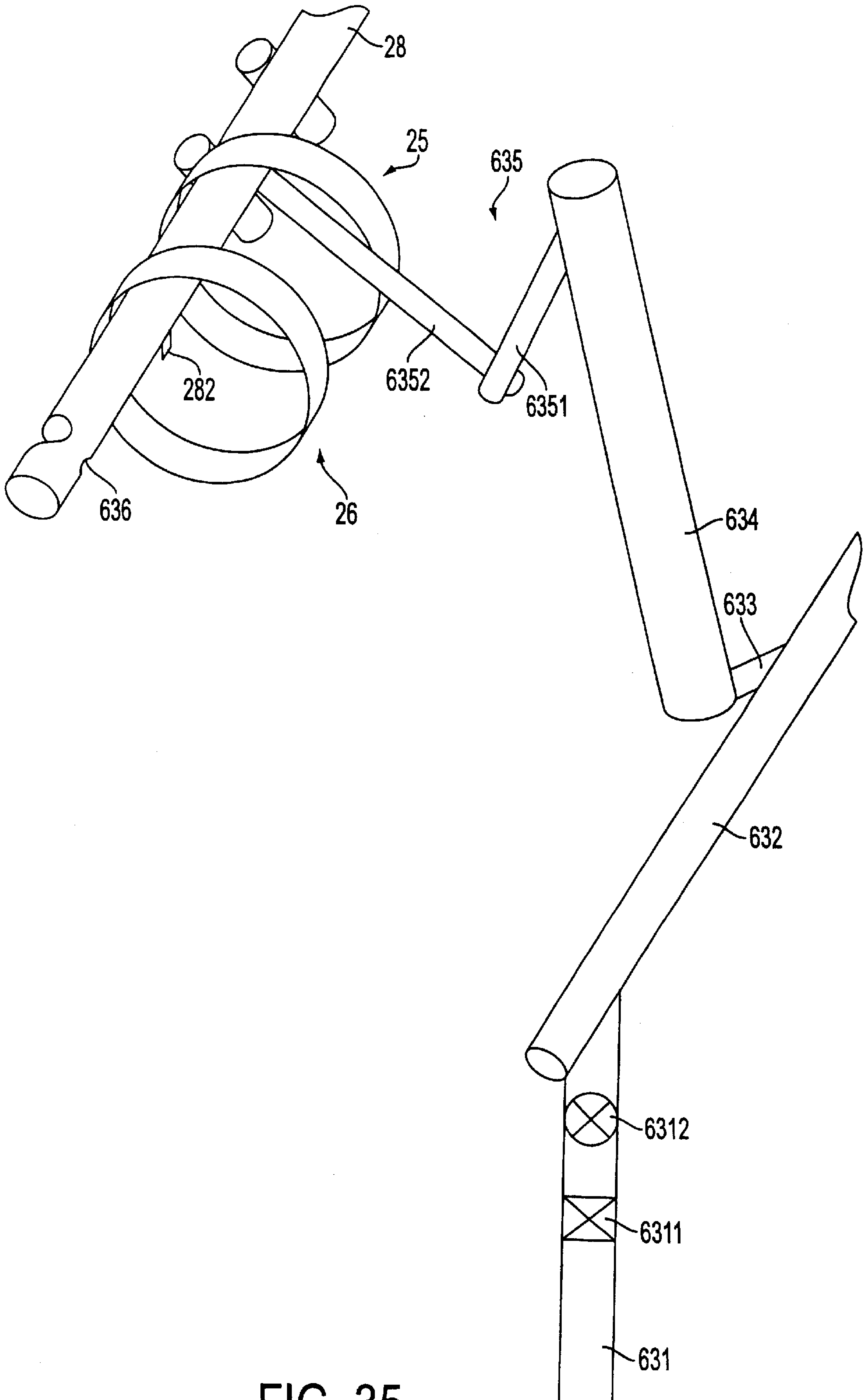

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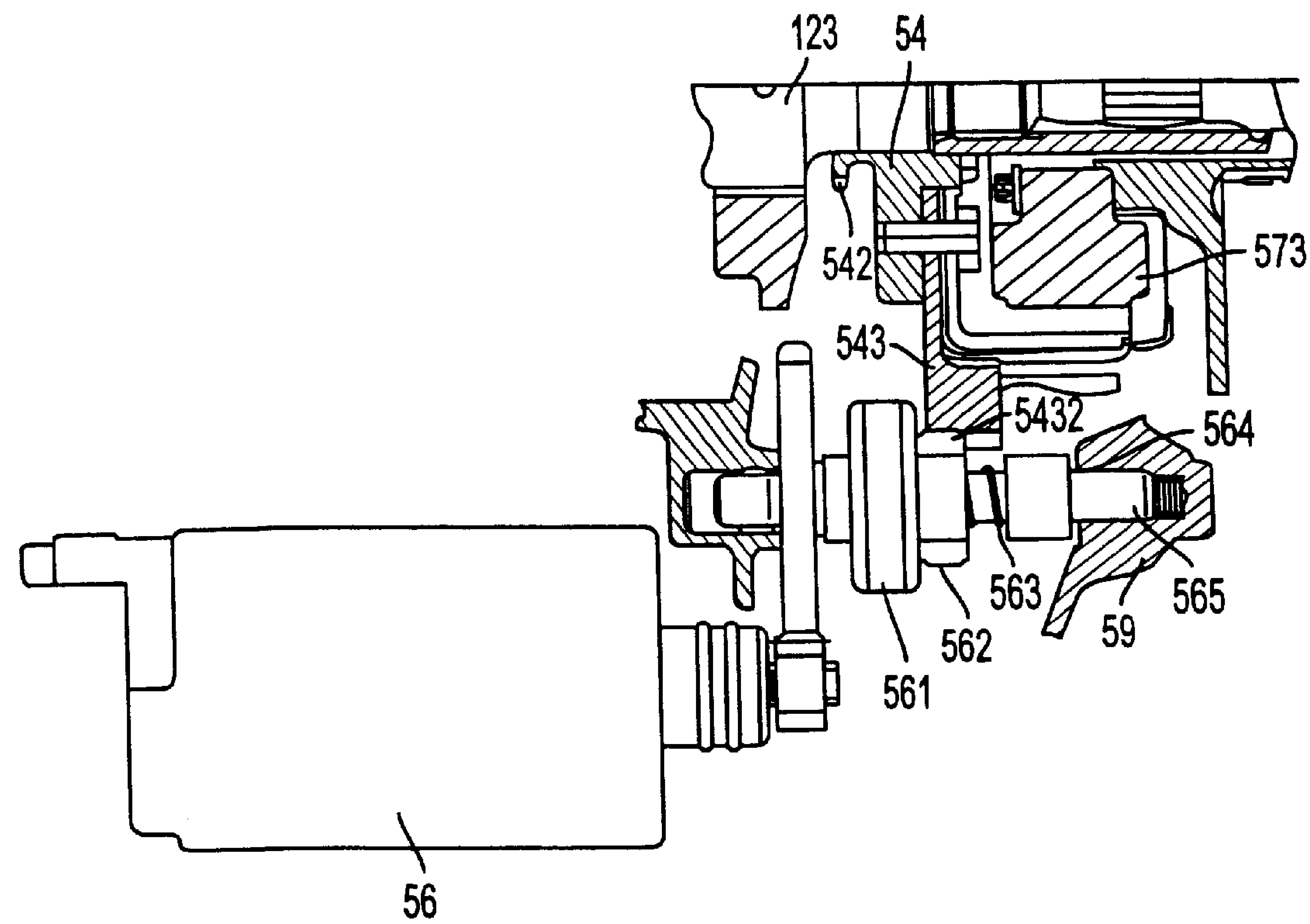

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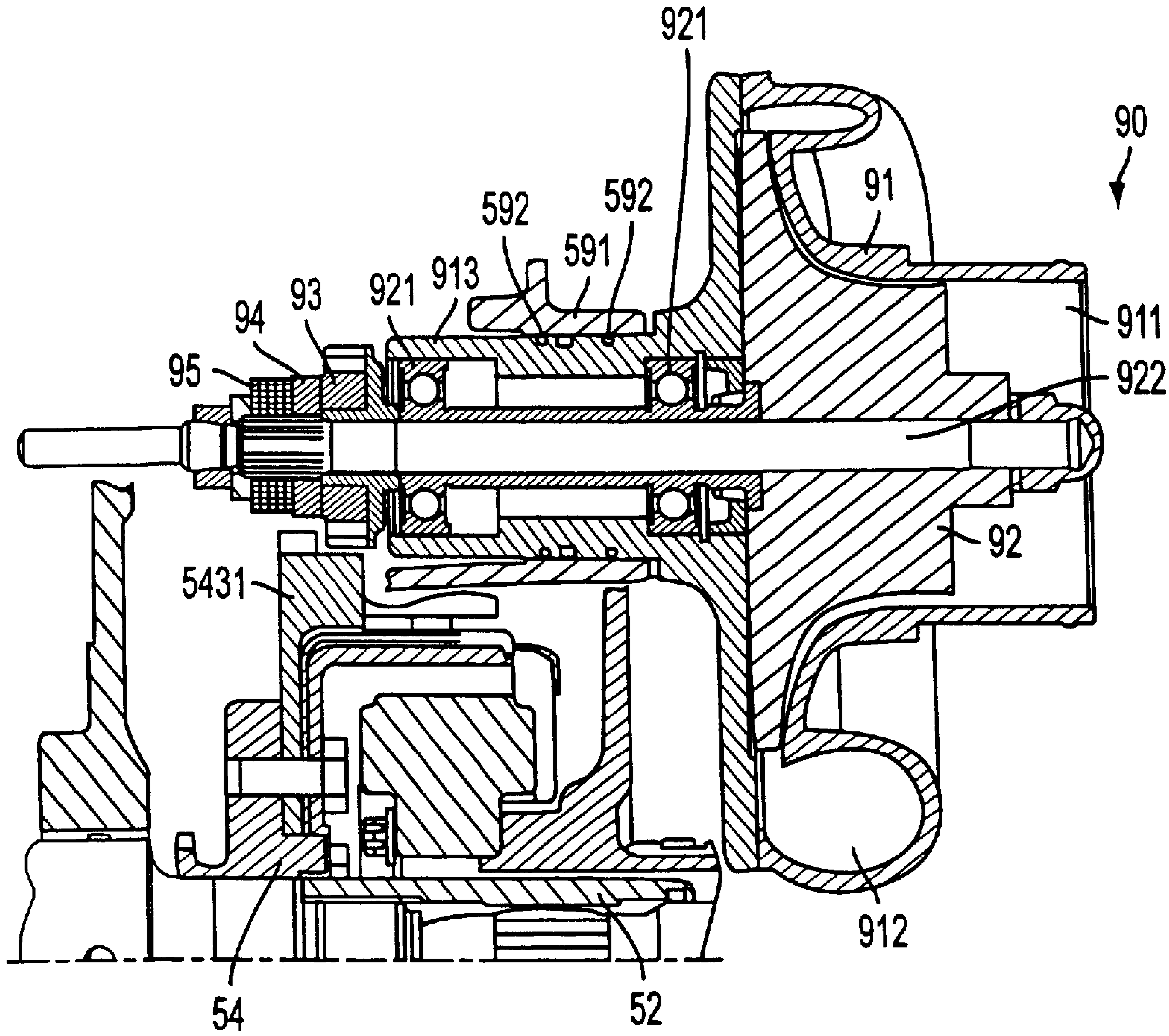

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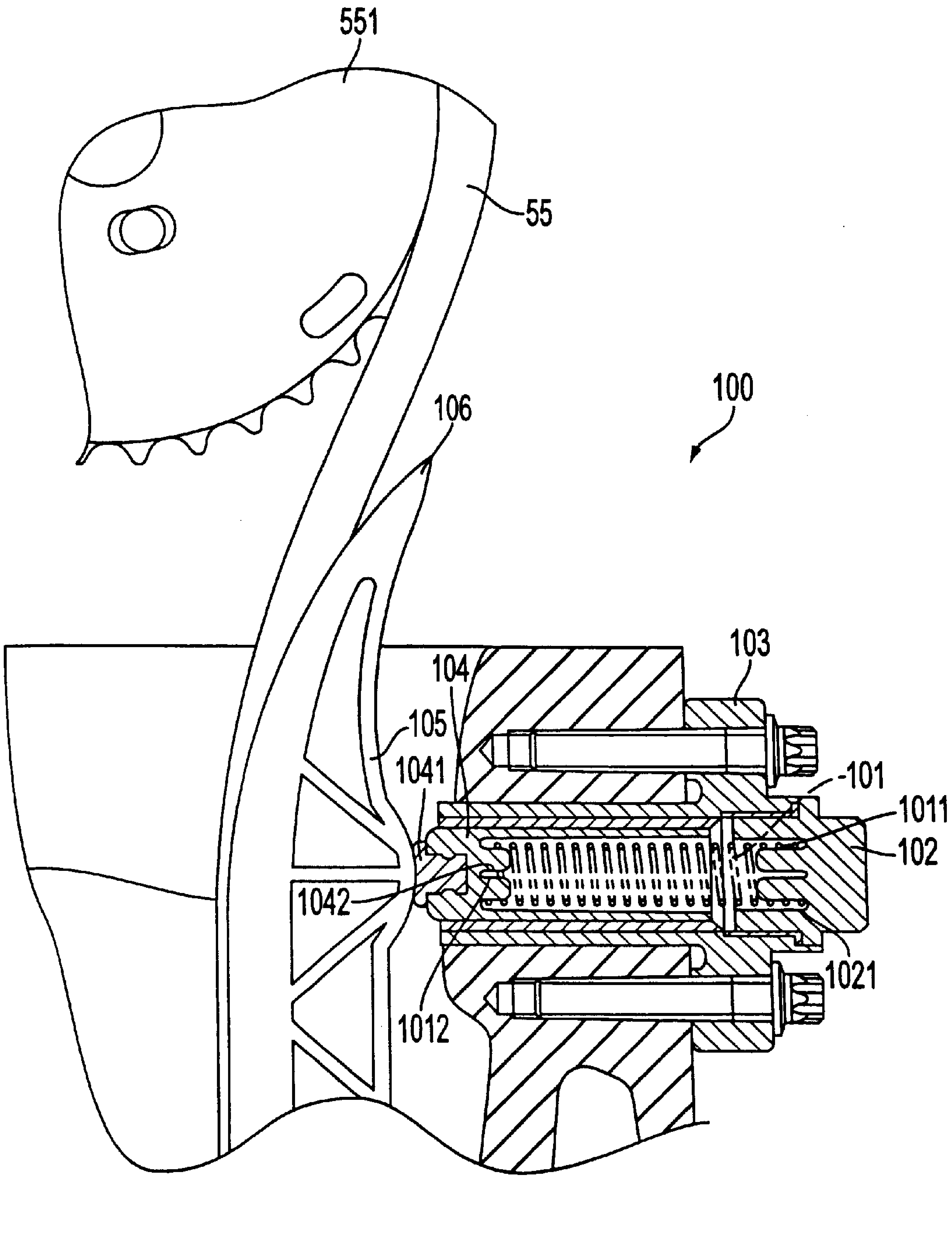

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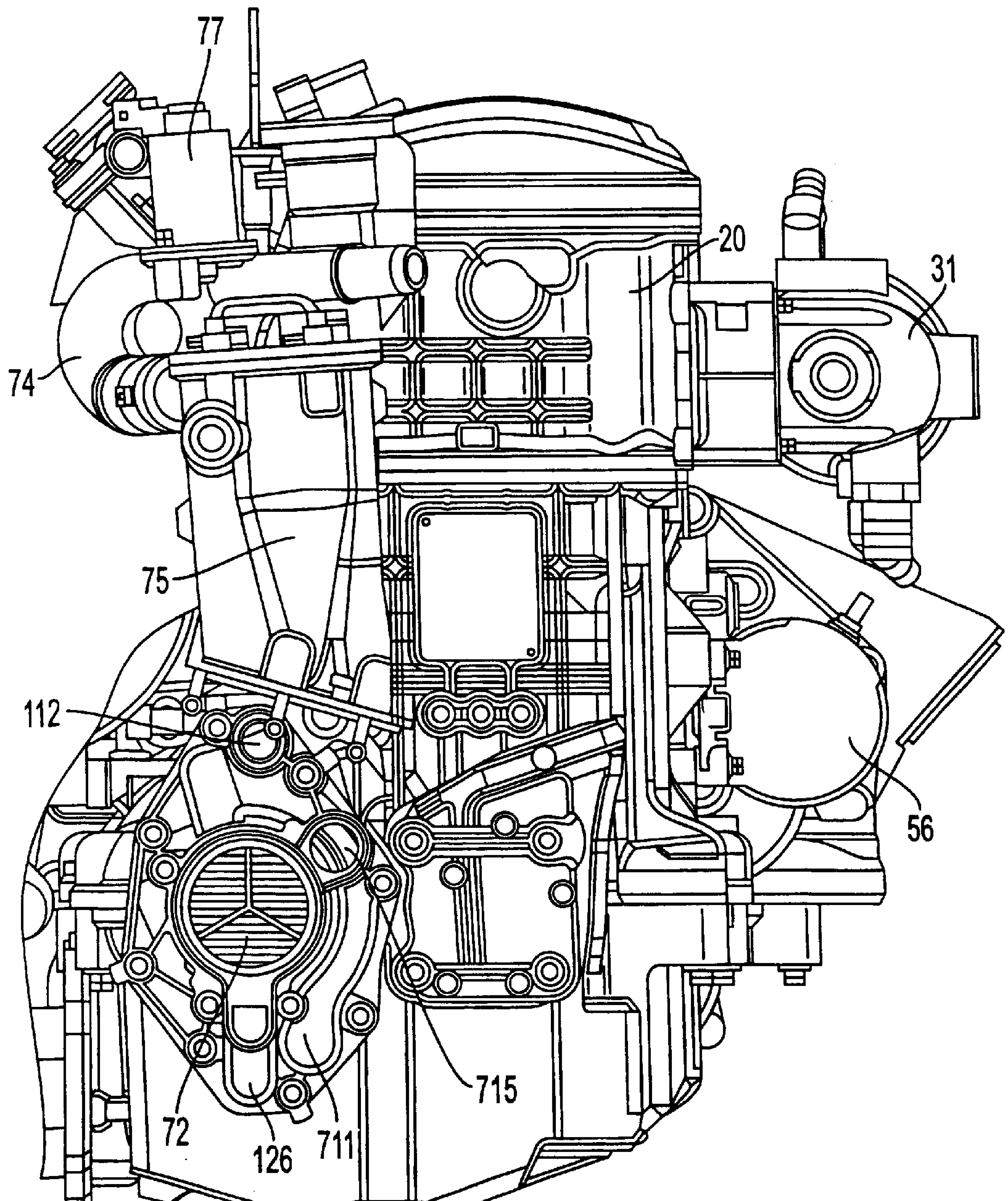

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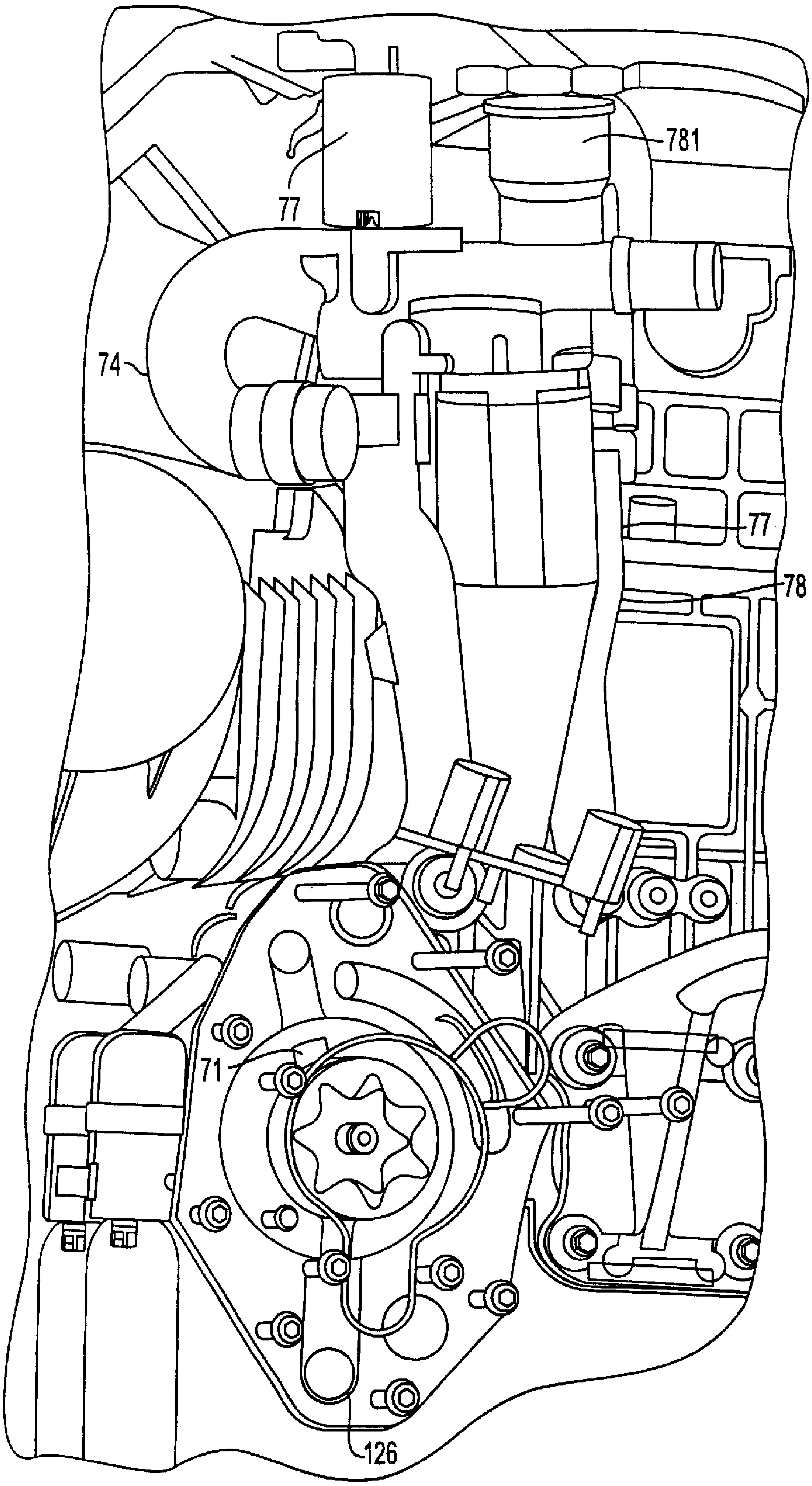

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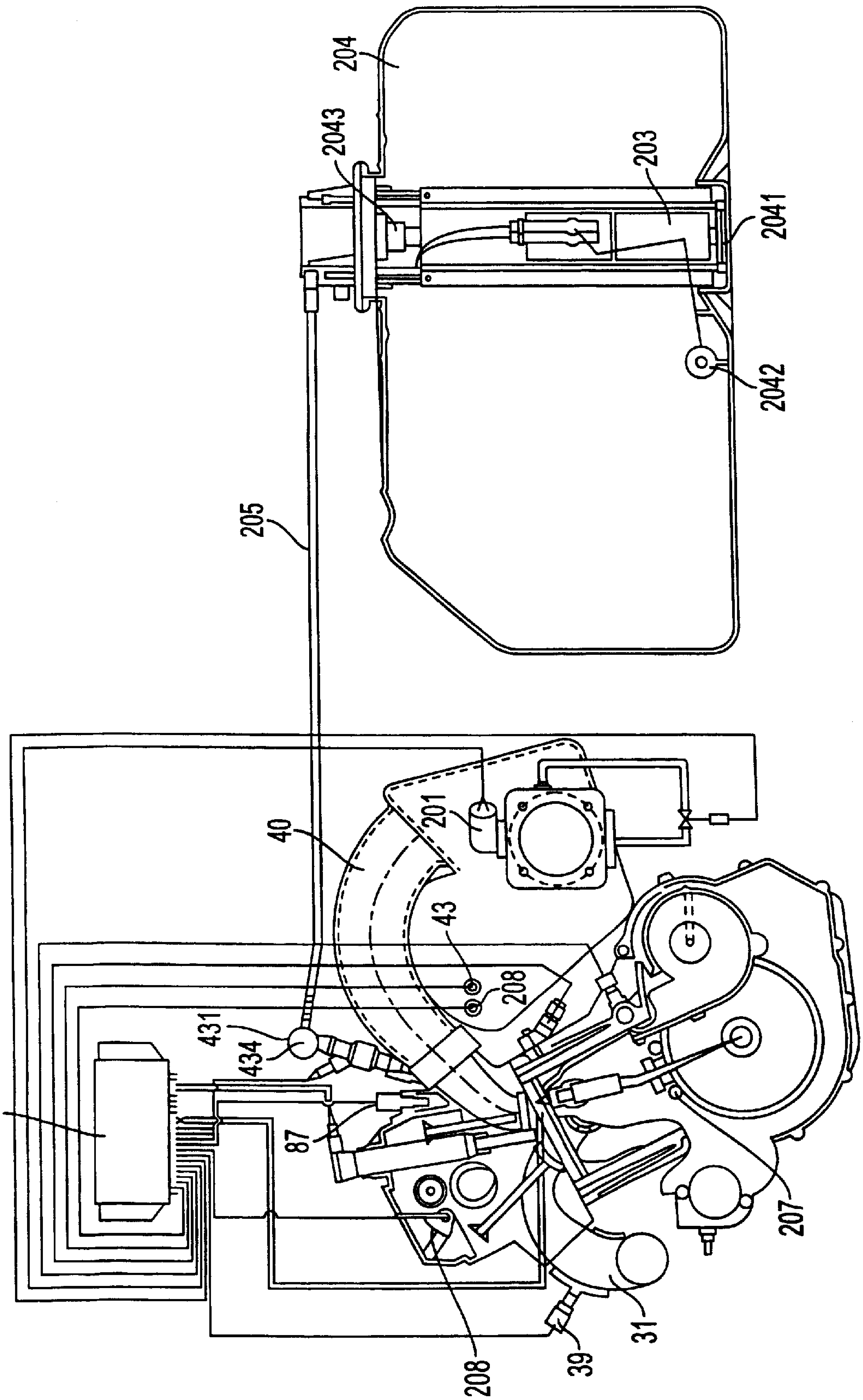
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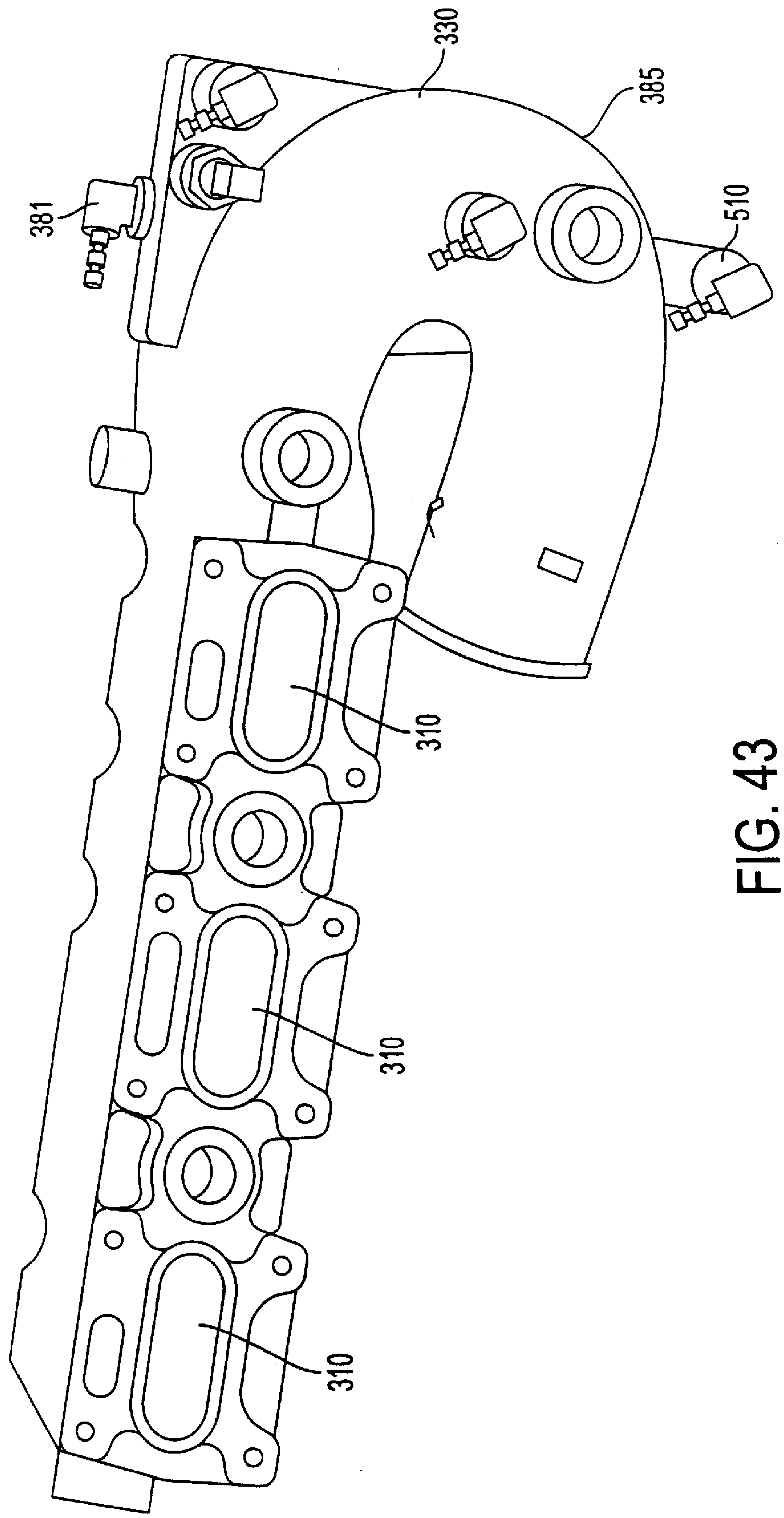
FIG. 42



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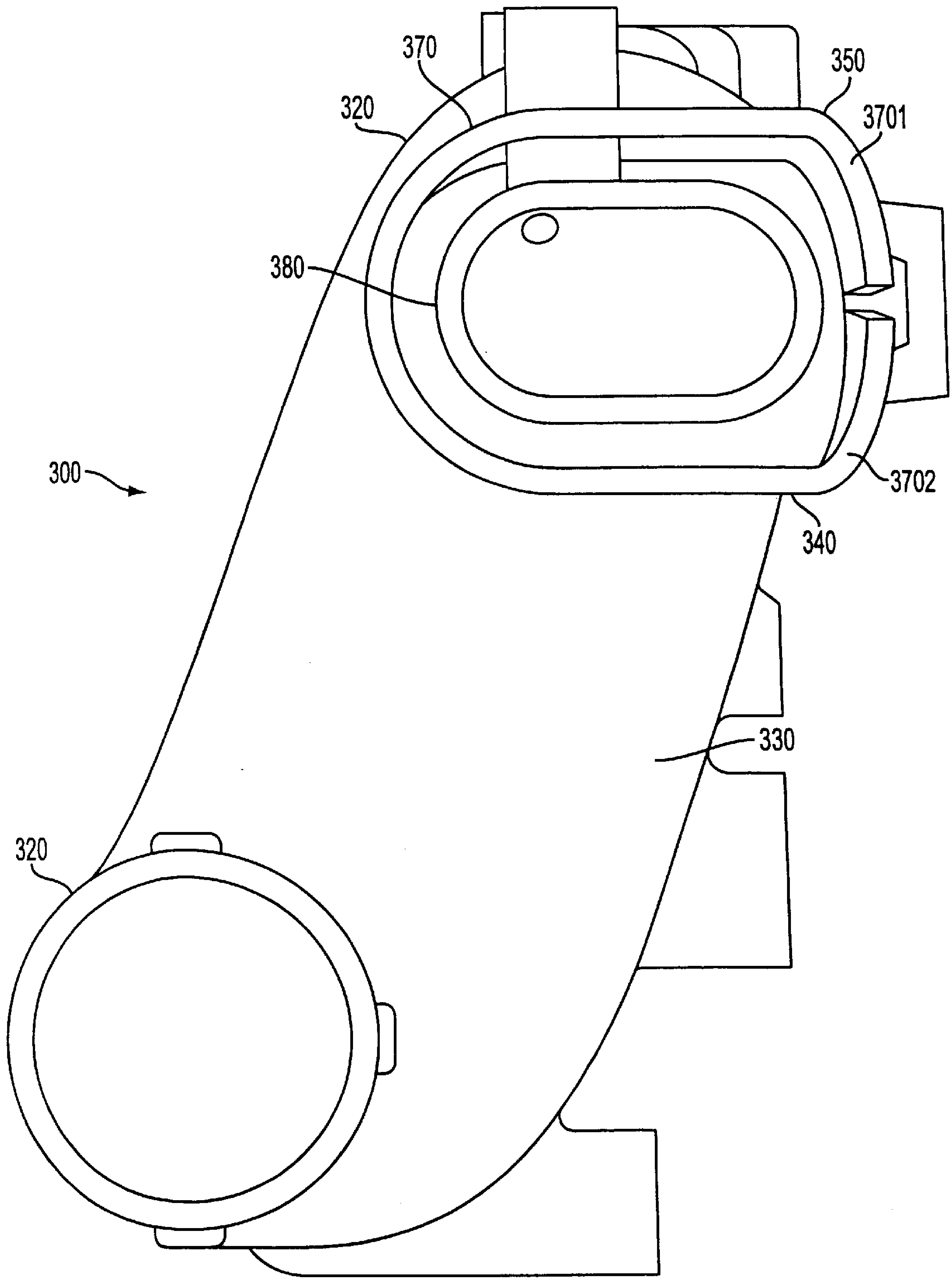

FIG. 44

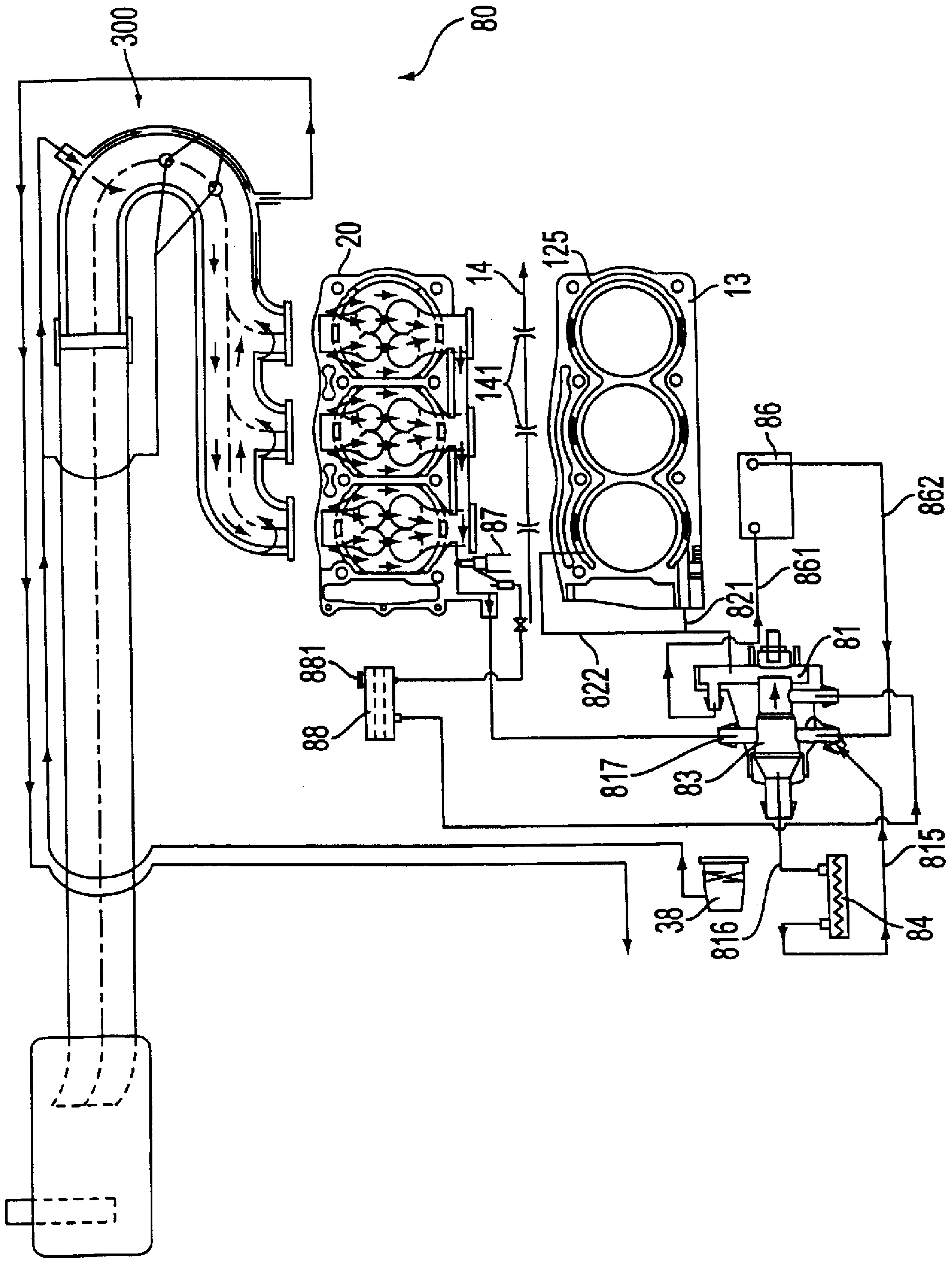
80
300
20
14
125
13
141
86
862
87
861
821
81
881
822
88
817
83
815
38
816
84

FIG. 45

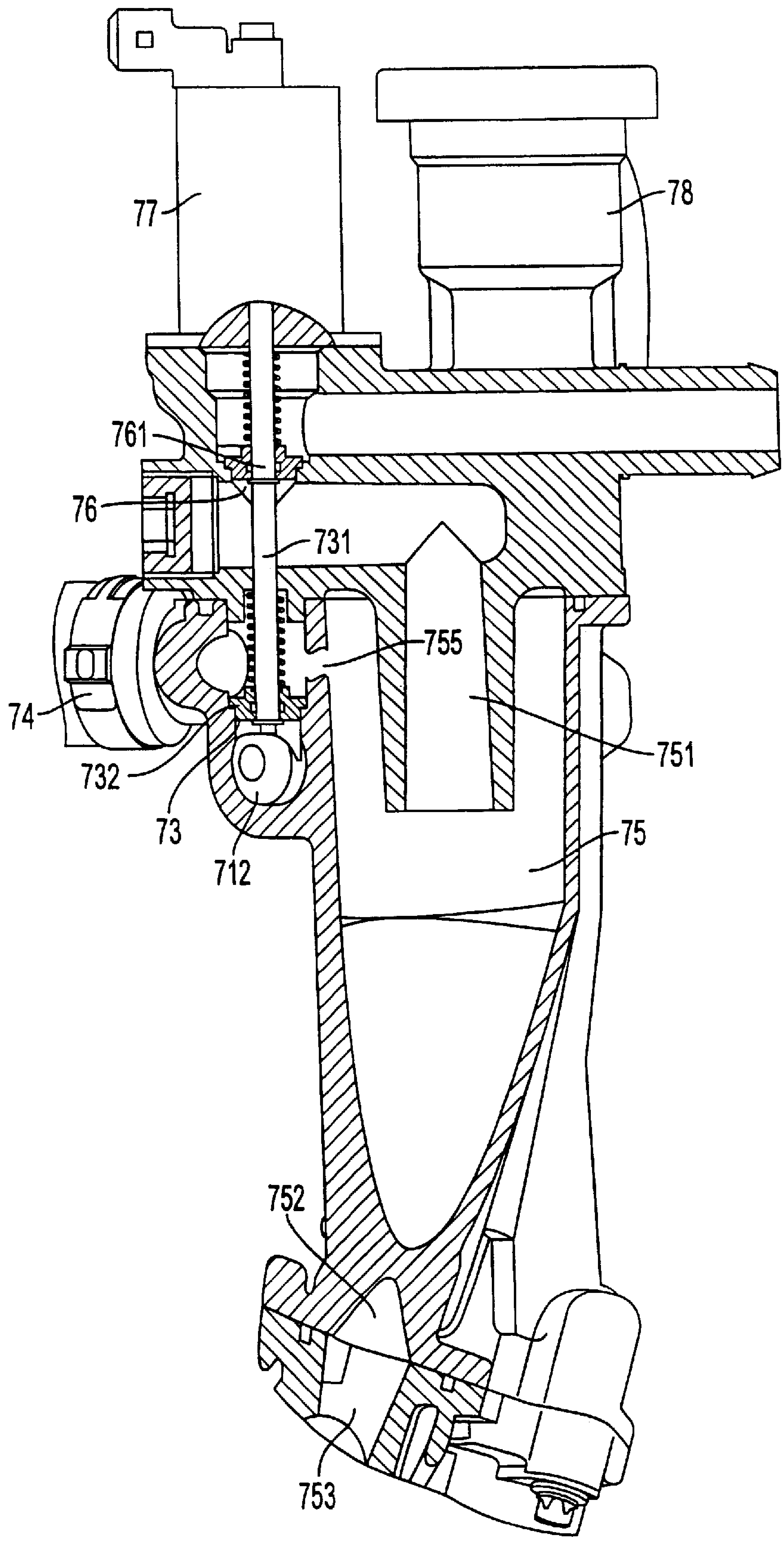

FIG. 46

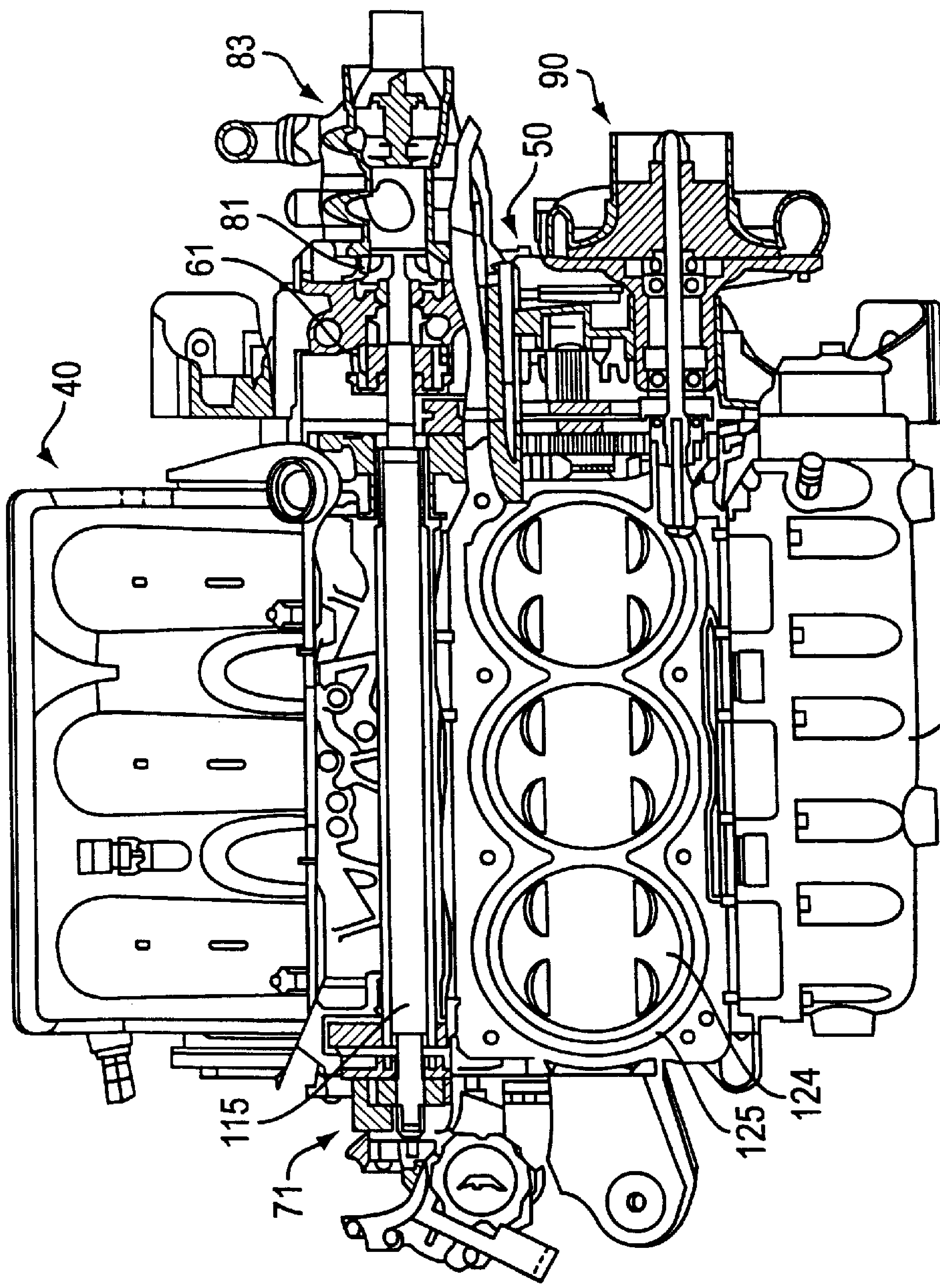

FIG. 47

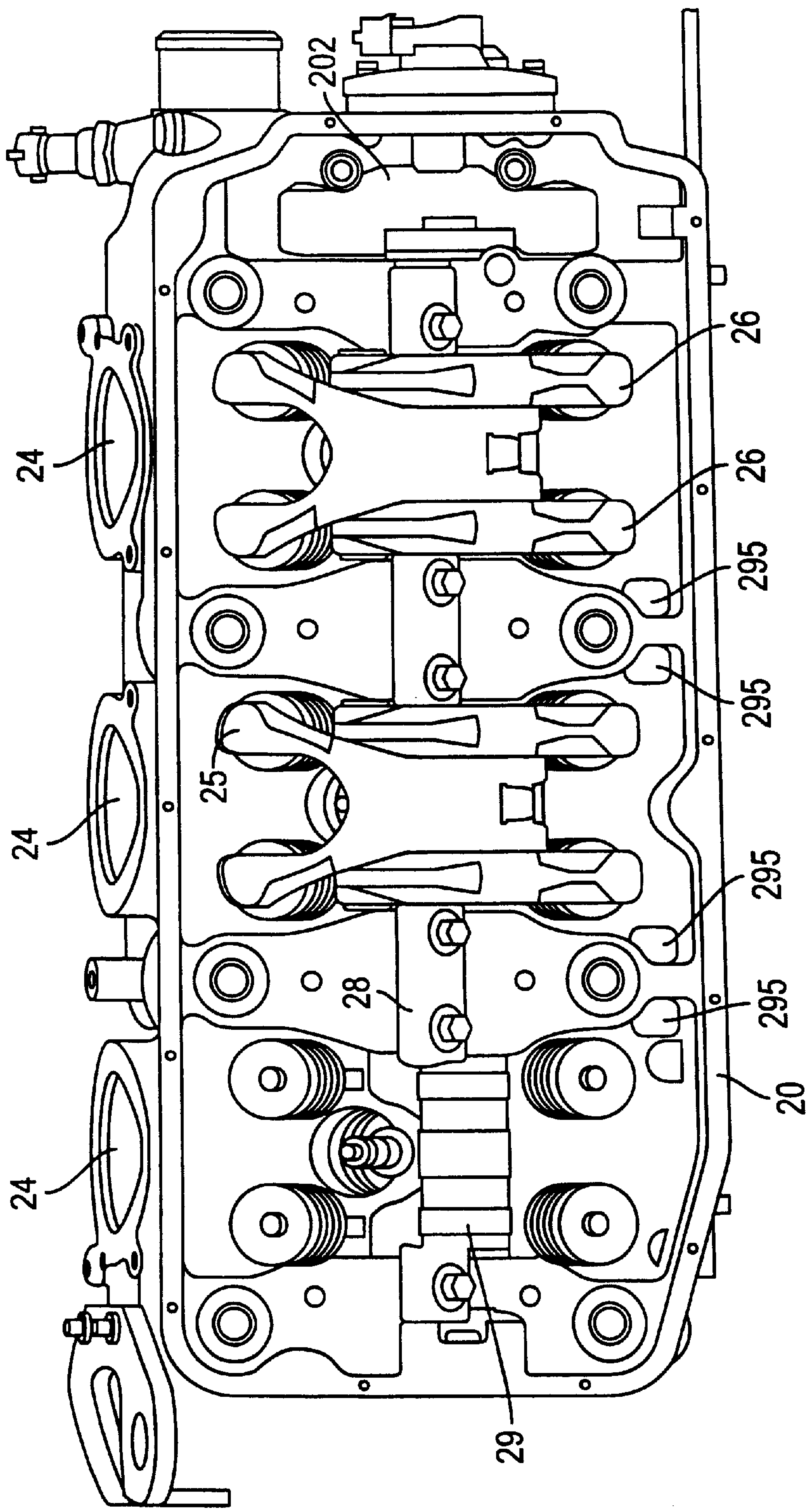

FIG. 48

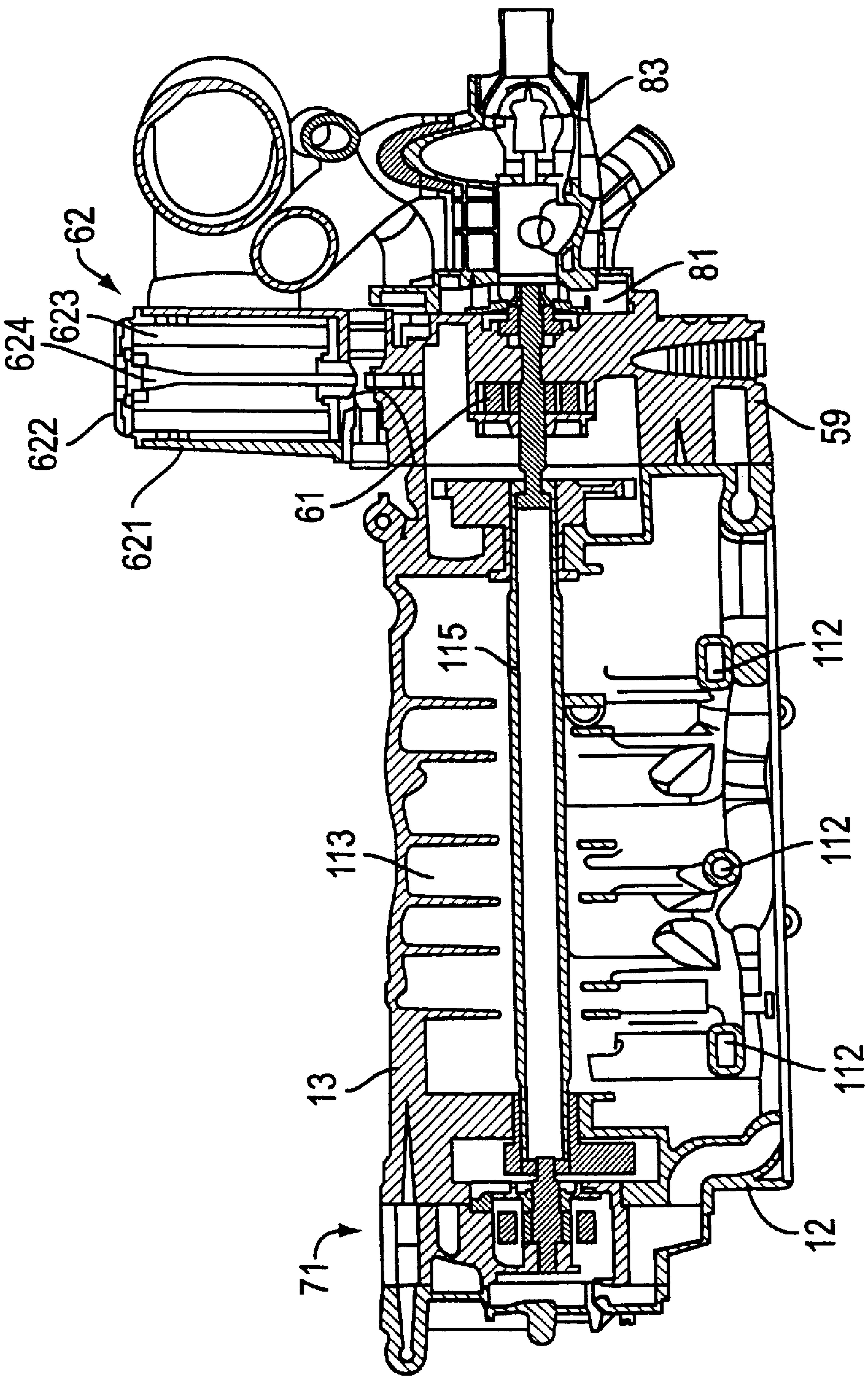

FIG. 49

FOUR STROKE ENGINE WITH VALVE TRAIN ARRANGEMENT

CROSS REFERENCE TO RELATED APPLICATIONS

This application relates to and claims priority on U.S. Provisional Application No. 60/185,703, filed on Feb. 29, 2000, and U.S. Provisional Application No. 60/257,174, filed on Dec. 22, 2000, which are incorporated by reference herein.

FIELD OF THE INVENTION

The present invention relates generally to a new engine for use in, for example, personal watercraft. In particular, the present invention relates to a new four-stroke in-line engine that was developed with a view to the future stricter environmental and emission regulations. The engine has an improved valve train arrangement for a lower engine profile and easy access to the spark plug assembly.

BACKGROUND OF THE INVENTION

There is a very popular type of watercraft known as a "personal watercraft" which is designed to be operated primarily by a single rider. Although this type of watercraft is commonly employed for single riders, frequently provisions are made for accommodating additional passengers although the maximum number of passengers is more limited than conventional types of watercraft.

This type of watercraft is also generally quite sporting in nature and normally accommodates at least the rider on a type of seat in which the rider sits in a straddle fashion. The passenger's area is frequently open through the rear of the watercraft so as to facilitate entry and exit of the rider and passengers to the body of water in which the watercraft is operating, as this type of watercraft is normally ridden with passengers that are wearing swimming suits.

These personal watercraft are generally quite small so that they can be conveniently transported from the owner's home to a body of water for its use. Because of the small size, the layout of the components is extremely critical, and this gives rise to several design considerations that are peculiar to this type of watercraft. However, due to its sporting nature it is also desirable if the watercraft is powered by an engine and propulsion device that are not only efficient but also generate sufficient power.

Traditionally, two-cycle engines have been used to power watercraft, including personal watercraft. These engines have the advantage that they are fairly powerful, relatively lightweight, and compact.

One particular disadvantage to the two-cycle engine is its emission content. Two-cycle engines generally exhaust larger quantities of hydrocarbons and other pollutants than four-cycle engines due to cylinder charging inefficiencies and the combustion of lubricating oil among other things. When measures are taken to reduce emissions of the two-cycle engine, other generally undesirable consequences can result, such as an increase in the weight of the engine, a reduction of its power output or the like. With concern for the environment and increasingly strict emissions requirements being instituted by various governing bodies. There is motivation to provide a power plant that reduces exhaust emissions while retaining other advantageous characteristics such as compactness, low weight and high power output.

Four-cycle engines are commonly used as power plants in other applications, such as automobiles. These engines have the advantage that their emissions output are generally desirably lower as compared to a two-cycle engine for a given power output. These engines, however, are considerably larger than two-cycle engines and therefor present difficulties when locating the engine in a personal watercraft. It is desirable to provide an engine with a reduced profile. This may be accomplished with proper configuration of the valve train arrangement and cylinder head.

U.S. Pat. No. 4,267,811 to Springer, entitled "Cylinder Head For a Mixture-Compressing Internal Combustion Engine," U.S. Pat. No. 4,553,515 to King et al., entitled "Cylinder Head For Spark Ignition Internal Combustion Engine," U.S. Pat. No. 4,662,323 to Moriya, entitled "Overhead Cam Type Valve Actuating Apparatus For Internal Combustion Engine," U.S. Pat. No. 4,741,302 to Oda et al., entitled "Internal Combustion Engine," U.S. Pat. No. 4,773,361 to Toki et al., entitled "Overhead Cam Type Four-Valve Actuating Apparatus For Internal Combustion Engine," U.S. Pat. No. 4,796,574 to Fujii et al., entitled "SOHC Type Internal Combustion Engine," U.S. Pat. No. 5,009,204 to Ishii, entitled "Spark Plug Arrangement In An Overhead Camshaft Engine," U.S. Pat. No. 5,095,859 to Iwata et al., entitled "SOHC Type Internal Combustion Engine," U.S. Pat. No. 5,513,606 to Shibata, entitled "Marine Propulsion Unit," U.S. Pat. No. 5,829,402 to Takahashi et al., entitled "Induction System For Engine," U.S. Pat. No. 5,839,930 to Nanami et al., entitled "Engine Lubricating System For Watercraft," and U.S. Pat. No. 5,846,102 to Nitta et al., entitled "Four-Cycle Engine For A Small Jet Boat" disclose various valve train arrangements for an internal combustion engine. Each discloses positioning the intake and exhaust valves at an angle within the cylinder head. Rocker arm assemblies are used to actuate the valves. None of these references, however, disclose using a single rocker arm having a pair of operating arms to operate a pair of valves such that the spark plug assembly is located between the operating arms of the rocker arm. Furthermore, none of these references provides a spark plug assembly that permits easy removal of the spark plug while protecting the same from the elements.

OBJECTS OF THE INVENTION

It is an object of the present invention to provide a four stroke, in-line engine having a compact construction.

It is another object of the present invention to provide a four stroke, in-line engine having a modular construction to permit the interchange of parts between various engine models.

It is another object of the present invention to provide a four stroke, in-line engine having improved exhaust emission characteristics.

It is another object of the present invention to provide a four stroke engine having a narrow and low profile.

It is another object of the present invention to provide a four stroke engine having a low profile valve actuation assembly for controlling the operation of the intake and exhaust valves.

It is another object of the present invention to provide a cylinder head having a low profile to reduce engine height.

It is another object of the present invention to offset the placement of the intake valves and exhaust valves with respect to a vertical axis within the cylinder head to reduce engine height.

It is another object of the present invention to provide an improved spark plug mounting assembly for easy access within the cylinder head.

It is another object of the present invention to provide a Y-shaped intake rocker arm assembly providing compact construction.

It is yet another object of the present invention to provide a four stroke engine having an improved oil collection system and oil holding tank.

It is another object to provide a four stroke engine which combines a closed loop cooling system and an open loop cooling system for enhanced cooling of the engine in accordance with the present invention.

It is another object to provide an open loop cooling system for cooling an exhaust manifold in accordance with the present invention, wherein the open loop cooling system enhances cooling of the crankcase and cylinder head.

It is another object to provide an open loop cooling system for cooling an exhaust manifold in accordance with the present invention, wherein the open cooling system lowers the temperature of the exhaust manifold such that the exhaust manifold functions as a heat sink for the crankcase and cylinder head.

It is another object of the present invention to provide a closed loop cooling system for selectively cooling the crankcase and cylinder head of the four stroke engine.

It is another object of the present invention to provide a closed loop cooling system having a selectively operable heat exchanger.

It is another object of the present invention to provide a supercharger for enhanced engine performance.

SUMMARY OF THE INVENTION

In accordance with the present invention, a four stroke internal combustion engine is disclosed. In the preferred form, the engine includes a crankcase having a crank shaft rotatably mounted therein. A cylinder head is connected to the crankcase. The crankcase and the cylinder head together form at least one cylinder. Each cylinder has at least one intake valve and at least one exhaust valve a crankshaft rotatably mounted within the crankcase. A valve actuation assembly operates the intake valves and the exhaust valves. The valve actuation assembly is located in the cylinder head between the intake valve axis and the exhaust valve axis.

In accordance with the present invention, the valve actuation assembly includes a cam shaft that is rotatably mounted within the cylinder head. It is preferable that the cam shaft is operatively coupled to the crank shaft such that rotational movement of the crankshaft is transferred to the cam shaft. A support axle is mounted within the cylinder head and offset from the cam shaft. The support axle has a central passageway. The valve actuation assembly includes at least one exhaust rocker arm is pivotally mounted within the cylinder head on the support axle. Each exhaust rocker arm is operatively coupled to the cam shaft for operating the exhaust valves. The valve actuation assembly also includes at least one intake rocker arm is also pivotally mounted within the cylinder head on the support axle. Each intake rocker arm is operatively coupled to the cam shaft for operating the intake valves.

In accordance with the present invention, the engine further includes a lubrication system for lubricating the engine. The lubrication system includes a supply of lubricant to the cylinder head. A portion of the supply of lubricant flows through the central passageway in the support axle.

In order to reduce the overall profile of the engine and provide sufficient space a spark plug assembly and the necessary crank gear to cam gear ratio, the intake valves and exhaust valves are disposed at an angle with respect to the longitudinal axis of the cylinder. To accomplish this, the cam shaft and the support axle are also offset with respect to the longitudinal axis. At least one of the cam shaft and the support axle are positioned closer to the exhaust valves than the intake valves. It is contemplated that the longitudinal axis of the spark plug assembly extends substantially parallel to the longitudinal axis of the intake valves. It is also contemplated that the longitudinal axis of the spark plug assembly and the intake valve axis may form an acute angle not greater than 20°.

In accordance with the present invention, each exhaust rocker arm includes a cam follower located on one end of the exhaust rocker arm. The cam follower is adapted to follow a profile of an exhaust cam lobe located on the camshaft to operate the exhaust valves. An exhaust hydraulic adjuster located on an opposite end of the exhaust rocker arm. Each exhaust rocker adjuster includes a slidable piston assembly that is adapted to contact and operate the exhaust valve in response to movement of the exhaust rocker arm by the exhaust cam lobe. A fluid passageway extends from the piston assembly through the exhaust rocker arm to the support axle. Fluid in the central passageway in the support axle flows through the fluid passageway to bias the piston assembly into engagement with the exhaust valve.

In accordance with the present invention, the valve actuation assembly includes an intake rocker arm for operating a pair of intake valves. Like the exhaust rocker arm, each intake rocker arm includes a cam follower located on one end of the intake rocker arm. Unlike the exhaust rocker arm, the intake rocker arm includes a first actuator arm for operating a first intake valve, and a second actuator arm for operating a second intake valve. Each actuator arm includes a slidable piston assembly, as described above. The cam follower and actuator arms have a generally Y-shape.

In accordance with the present invention, the exhaust rocker arms are rotatably mounted on the support axle on opposite sides of the intake rocker arm. A spark plug assembly is positioned on the cylinder head between the first actuator arm and the second actuator arm.

The spark plug assembly includes a tube assembly secured to the cylinder head. The tube assembly may be plastic. A spark plug connector is removably located within the tube assembly. The spark plug connector may include a splash cover to prevent contaminants from entering the spark plug assembly. A spark plug is secured to the spark plug connector.

The present invention is also directed to a personal watercraft for at least one passenger having an internal combustion engine secured to the hull below a seating assembly. The internal combustion engine includes a valve actuation assembly, as described above and described in greater detail below.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be described in conjunction with the following drawings in which like reference numerals designate like elements and wherein:

FIG. **1** is a downward rear schematic perspective view of a left side of an overhead camshaft aspirated engine in accordance with the present invention;

FIG. **2** is a downward rear schematic perspective view of a right side of the engine of FIG. **1**;

FIG. **3** is a downward front schematic perspective view of the left side of the engine of FIG. **1**;

FIG. **4** is a downward front schematic perspective view of the right side of the engine of FIG. **1**;

FIG. **5** is a rear end view of the engine of FIG. **1** illustrating one possible positioning of the engine within a personal watercraft;

FIG. **6** is a downward rear schematic perspective view of a variation of the engine of FIG. **1** having a supercharger;

FIG. **7** is a rear end view of the engine of FIG. **6**;

FIG. **8** is a partial cross-sectional end view of the crankcase and cylinder head housing in accordance with the present invention;

FIG. **9** is a bottom view illustrating the upper crankcase of the engine in accordance with the present invention;

FIG. **10** is a top view of the lower crankshaft illustrating the positioning of the crankshaft and the balance shaft;

FIG. **11** is a right side partial schematic sectional view of the engine of FIG. **6**;

FIG. **12** is a partial schematic sectional view of the piston, valves and valve actuator assembly in accordance with the present invention;

FIG. **13** is a partial overhead schematic view of the rocker arm assemblies of the valve operating assembly for operating the intake and exhaust valves;

FIG. **14** is an end cross sectional view of one of the exhaust rocker arm assemblies and a portion of the intake rocker arm assembly taken along section line **14**—**14** of FIG. **13**;

FIG. **15** is a cross sectional view of the operative end of the rocker arm assemblies showing a collapsed position of the hydraulic adjuster on the left side and an extended position of the hydraulic adjuster on the right side;

FIG. **16** is a right side cross sectional view of the valve operating assembly located within the cylinder head having the camshaft in cross section;

FIG. **17** is another right side cross sectional view of the valve operating assembly located within the cylinder head;

FIG. **18** is an end cross sectional view illustrating the spark plug assembly within the cylinder head;

FIG. **19** is a cross sectional view illustrating the placement of the cylinder head cover on the cylinder head;

FIG. **20** is a cross sectional view of the engine of FIG. **1** through one cylinder of the engine;

FIG. **21** is a schematic perspective view of the exhaust manifold in accordance with the present invention;

FIG. **22** is a longitudinal cross sectional view of a portion of the exhaust manifold of FIG. **21**;

FIG. **23** is a side cross sectional view of a portion of the exhaust manifold of FIG. **21**;

FIG. **24** is a schematic view of the exhaust manifold and open loop cooling system in accordance with the present invention;

FIG. **25** is a schematic diagram of the cooling system for the engine in accordance with the present invention;

FIG. **26** is a rear perspective view of a right side of the air intake and fuel injection system for the engine in accordance with the present invention;

FIG. **27** is a cross sectional view of the air intake and fuel injection system of FIG. **26** taken along a longitudinal axis of the system;

FIG. **28** is a side cross sectional view of the air intake and fuel injection system of FIG. **26** through a swing pipe;

FIG. **29** is a variation of the air intake and fuel injection system of FIG. **28** illustrating a cooling jacket within the swing pipe;

FIG. **30** is a front perspective view of a right side of the air intake and fuel injection system for the engine having a supercharger in accordance with the present invention;

FIG. **31** is a cross sectional view of the air intake and fuel injection system of FIG. **30** taken along a longitudinal axis of the system;

FIG. **32** is a rear view of the engine illustrating the power take off lid and cooling system in accordance with the present invention and the oil filter housing in partial cross section;

FIG. **33** is a side cross sectional view of a thermostat and pump assembly of a portion of the cooling system and a lubrication pump of the lubrication assembly in accordance with the present invention;

FIG. **34** is a partial schematic/partial side cross sectional view of an oil filter unit in accordance with the present invention;

FIG. **35** is a schematic diagram illustrating the oil channel system for the lubrication system for the cylinder head housing;

FIG. **36** is a cross sectional side view of the power take off assembly for the engine illustrating the generator assembly in accordance with the present invention;

FIG. **37** is another cross sectional side view of the power take off assembly for the engine illustrating the starter assembly in accordance with the present invention;

FIG. **38** is a cross sectional side view of the power take off assembly having a supercharger for the engine in accordance with the present invention;

FIG. **39** is a partial schematic/partial sectional view of the cam chain tensioner in accordance with the present invention;

FIG. **40** is a schematic view of the blow-by ventilation system and suction pump in accordance with the present invention;

FIG. **41** is a schematic view of the blow-by ventilation system and suction pump of FIG. **38** having the suction pump cover removed;

FIG. **42** is a schematic view of the engine management system for the engine in accordance with the present invention;

FIG. **43** is a schematic perspective view of the exhaust manifold according to an alternative embodiment;

FIG. **44** is a cross sectional view of a portion of the exhaust manifold of FIG. **43**;

FIG. **45** is a schematic diagram of the cooling system for the engine in accordance with the present invention for use in connection with the exhaust manifold of FIG. **43**;

FIG. **46** is a cross sectional view of the cyclone of the blow-by ventilation system;

FIG. **47** is a partial overhead cross sectional view of the engine of FIG. **6** having a cut away of the balance shaft and the power take off assembly;

FIG. **48** is an overhead view of the valve train;

FIG. **49** is a partial side cross sectional view of the balance shaft and power take off assembly; and

FIG. **50** is a side view of the engine of FIG. **1** illustrating one possible positioning of the engine within a personal watercraft.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

A four-stroke three cylinder in-line engine **1** in accordance with the present invention is illustrated generally in

FIGS. 1–4. The engine **1** in accordance with the present invention will be described in connection with a personal watercraft **5**, shown in cross-section in FIG. **5**. A variation of the engine **1** is illustrated in FIGS. **6** and **7**. The engine **2** shown in FIGS. **6** and **7** includes a supercharger. The engines **1** and **2** are adapted to be installed below a raised pedestal having a seating bench of the personal watercraft **5** inside the hull **4**, as shown in FIGS. **5** and **50**. With this arrangement, the oil filter cannot be placed on the lower side of the engine or of its crankcase, respectively, if it is to be accessible for maintenance purposes because the hull **4** would prevent access to the oil filter. To address this, the oil filter is installed at the power take off side of the engine, to be easily accessible from above. The access through the seating area at present is the only access to the engine.

While designed for use in personal watercraft, it is contemplated that the engine **1** (or engine **2**) can be used in all terrain vehicles, snowmobiles, boats and other vehicles with minor modifications. For example, the cooling system for the exhaust manifold must be modified for non-marine applications. Further, while the embodiments shown disclose an engine positioning with the power take off to the rear of the engine, the orientation can be altered to have the power take off to the front or to the side depending on the specific vehicle or specific application.

Engine Configuration

The engine **1** includes a crankcase **10**. A cylinder head housing **20** is connected to the crankcase **10** to form a plurality of combustion chambers. The crankcase **10** and cylinder head housing **20** are inclined with respect to a vertical axis, as shown in FIGS. **5** and **8**. This arrangement provides sufficient space for the air intake and fuel injection system **40** while maintaining an overall reduced engine profile. The engines illustrated and described herein include three cylinders. The present invention, however, is not limited to three cylinders; rather, it is contemplated that a greater or fewer number of cylinders are considered to be well within the scope of the present invention. For example, a single cylinder version of the engine may be employed in a fishing boat. Two or three cylinder versions of the engine may be employed in a personal watercraft. A four cylinder version of the engine may be employed in a jet boat. Four or more cylinders are considered to be well within the scope of the present invention.

The engine **1** or **2** provides for the location of various engine components including, but not limited to the starter assembly, the generator, the oil pump, coolant pump and other devices at one end of the engine in the power take off assembly **50**, described below and shown in FIGS. **33, 36, 37** and **38**. This unique construction and layout of components permits the use of similar parts and engine components for one, two, three and four cylinder versions of the engine. Furthermore, this arrangement permits the addition of additional cylinders on the end of the engine opposite the power take off assembly. The layout of the parts is the same. Minimal redesign of these components is necessary when increasing or reducing the number of cylinders.

The engine **1** contemplated herein includes an exhaust manifold **30** that is secured to one side of the cylinder head housing **20** and an air intake and fuel injection system **40**. The air intake and fuel injection system **40** is secured to an opposite side of the cylinder head housing **20** in the area above the cylinder head housing **20**.

The present invention, however, is not limited to having a fuel injection system; rather, it is contemplated that the engine can instead be equipped with a carburetor.

A power take off assembly **50** is located on an end of the cylinder block **10** within the hull **4**. The power take off assembly **50** defines the rear side of the engine when located within the personal watercraft **5**. The engine **1** or **2** further includes a lubrication system **60** as shown in FIGS. **8** and **11**. The engine **1** further includes a blow-by ventilation system **70**, as shown in FIG. **11**, and an engine cooling system **80**, as shown in FIG. **25**.

An engine **2** is shown in FIGS. **6** and **7**, which is a variation of the engine **1**. The engine **2** has substantially the same configuration as the engine **1**. The engine **2** further includes a supercharger **90**. The use of a supercharger for an engine for use in a personal watercraft is a new development, which is described in greater detail below. The engine **1** can be converted with minor modification to the engine **2** having a supercharger **90**. In particular, as described below, the supercharger **90** is attached to an opposite end of the intake manifold **41** as compared to the normally aspirated engine **1**. The ignition and induction parameters of the engine may be modified to enhance engine performance when the supercharger **90** is used. It is also contemplated that the compression ratio of the engine may have to be altered to accommodate the supercharger **90**. In accordance with the present invention, it is contemplated that the engines **1** and **2** will be produced on the same assembly line.

Because it is contemplated that the engine in accordance with the present invention will be used in marine applications, the exterior surfaces of the engines **1** or **2** will be provided with a suitable coating to reduce corrosion and the direct exposure of the engine to the elements. The individual components of the engines **1** and **2** will now be described in greater detail.

Crankcase

As illustrated in FIG. **8**, the crankcase **10** contains a plurality of passageways and compartments formed therein. Furthermore, the crankcase **10** is formed with vertical partitions, as shown in FIGS. **9** and **10**, which separate the individual crank chambers, described below and external fins located on the crankcase **10**. These vertical partitions and external fins increase the strength of the crankcase **10**. The spaced apart vertical fins provide additional strength for an upper crankcase **13** of the crankcase **10** while minimizing the weight. The vertical partitions increase engine strength and separate the crank chambers **121** in the upper and lower crankcases **12** and **13**. The vertical partitions also secure the upper and lower crankcases together using suitable fasteners. The fasteners extend through bores in the vertical partitions from a lower end of the lower crankcase to the upper crankcase. The fasteners also serve to secure the bearings, described below, within the vertical partitions. The crankcase **10** is preferably formed from a cast aluminum alloy (e.g. AlSi) for both strength and weight considerations. The crankcase **10** is preferably die cast. The present invention, however, is not limited to the use of aluminum alloys; rather, other materials including but not limited to steels, alloys and composites are considered to be well within the scope of the present invention provided the materials have sufficient strength for use in engine applications.

The crankcase **10** includes an upper crankcase **13** containing the cylinder block and a lower crankcase **12**. A balance shaft **115** and a crankshaft **123** are located at the union between the lower crankcase **12** and the upper crankcase **13**. An oil tank **11** formed in a bottom portion of the lower crankcase **12**, as shown in FIG. **8**. The oil tank **11** has a generally u-shaped configuration that partially surrounds a lower portion of a crankcase **12**. The oil tank **11** is located on both the bottom and side of the engine to house the necessary volume of oil while maintaining the engine's reduced profile such that oil is located on the bottom of the crankcase and the e side of the crankcase **10**. An interior of the upper crankcase **13** and the lower crankcase **12** are connected to the oil tank **11** through outlet openings **111**, as shown in FIGS. **8** and **11**. A channel **112** extends from each opening **111** to an upper portion **113** formed in the lower crankcase **13**. The oil collected from the crank chamber **121** flows through outlet openings **111** and channels **112**, then enters the upper channel portion **113** and returns to the oil tank **11**. This oil then flows under the influence of gravity downward into a lower portion **114** of the oil tank **11**.

A balance shaft **115** extends through the crankcase **10**. The balance shaft **115** and the crankshaft **123** are located at the union of the lower crankcase **12** and the upper crankcase **13**. To prevent oil from flowing from upper channel portion **113** and contacting the balance shaft **115**, an optional baffle assembly is located within the upper portion **113**. The balance shaft **115** is provided to counteract the moment generated by rotation of the crankshaft **123**, shown in FIG. **10**. This arrangement produces mass balancing of the first order. The balance shaft **115** and the crankshaft **123** extend in a parallel relationship, as shown in FIG. **10**. The balance shaft **115** is rotatably mounted within a bore **1132** that extends through the crankcase **10**, as shown in FIGS. **9** and **10**. Suitable bearing assemblies are provided for smooth rotation of the balance shaft **115**. The bearing assemblies are fixed using the fasteners described above. Preferably, the balance shaft **115** should be mounted in an anti-friction shell bearing but, alternatively, roller bearings can also be used. The balance shaft **115** is operatively connected by gear **1151** to the crankshaft **123** through gear **1231**. This connection is preferably located within the power take off assembly **50** on one end of the crankcase **10**.

The oil tank **11** forms a portion of a dry sump lubrication system. The lubrication system and the operation of the same will be described in greater detail below.

As FIGS. **9** and **10** illustrate, the crankcase **10** includes at least one crank chamber **121** and in the preferred embodiment includes one isolated crank chamber for each engine cylinder. In accordance with the presently disclosed embodiments of engines **1** and **2**, three crank chambers **121** are provided. Each crank chamber **121** includes an outlet opening **111** connected to the channel **112**, described above. A bore **122** extends through the crankcase **10** and each of the crank chambers **121**, as shown in FIGS. **9** and **10**. A crankshaft **123** is received therein, as shown in FIG. **10**. The crankshaft **123** can be a one-piece forging, cast or assembled depending upon the engine application. For example, a cast crankshaft may be used in low performance applications. The crankshaft **123** is rotatably mounted within a bore **122**. Suitable bearing assemblies are provided for smooth rotation of the crankshaft **123**.

As shown in FIG. **25**, a cylinder **124** extends through the crankcase **10** above each of the crank chambers **121**. In accordance with the present invention, the engines **1** and **2** each include three cylinders **124**, as shown in FIG. **11**. A piston **1241** is slidably received within the cylinder **124**. The piston **1241**, shown in FIG. **11**, reciprocates axially within the cylinder **124** as is known. The piston **1241** is connected to the crankshaft **123** through a connecting rod **1242** and piston pin **1243** to convert axial movement of the pistons **1241** to rotational movement of the crankshaft **123** and vice-versa. A cooling passageway **125** extends around the cylinders **124**, as shown in FIG. **25**. The cooling passageway **125** is connected to the engine cooling system **80** further described below. As shown in FIG. **25**, the cooling passageway **125** extends substantially around the perimeter of the cylinders. This passageway has a generally U-shaped configuration.

At present, the cylinder liners are formed with grey cast iron. The upper crankcase **13** is then cast around the liners. The upper crankcase **13** may be formed from under-eutectic AlSi (e.g. cast-AlSi **9**)(with 9% silicon). The interior of the cylinder liners may then be honed. The use of grey cast iron increases the weight of the crankcase **13**. It is desirable to eliminate the use of the cylinder liners. With this in mind, it is contemplated that the cylinder liners may be eliminated. Instead, an interior surface of the upper crankcase **13** can include a thermal coating to reduce friction. This coating may be applied plasma spraying or other suitable process. Alternatively, AlSi-alloys (alloys of aluminum and silicon) are used to form the liners for the cylinders **124**. The cylinder liners may be formed from over-eutectic AlSi with primary silicon grains therein (e.g. AlSi **19**)(with 19% silicon) to minimize friction and wear. The crankcase **10** may be formed from under-eutectic AlSi (e.g. cast-AlSi **9**)(with 9% silicon). The cylinder liners are assembled to the cylinder block during the casting of the upper crankcase **13**. Beforehand, a binding layer consisting of eutectic AlSi **12** (with 12% silicon) is thermally sprayed (e.g. plasma sprayed) onto the outer wall of the liner to provide a better bond and a better heat-removal property (high heat transfer coefficient) between the liner and the cylinder block **10**. Alternatively, the cylinder liners can also be inserted into the cylinder block of the upper crankcase **13** mechanically with a force fit. It is also contemplated that the cylinder block **10** can be formed from over-eutectic AlSi (e.g. AlSi **19**) without the need for separate cylinder liners. With this arrangement, however, the cylinder is more difficult to machine, more expensive and thus, is not presently preferred. In such a liner-less embodiment, the cylinders can be optionally provided with a surface coating for enhanced wear and friction properties. It is contemplated that the pistons **1241** may be formed of aluminum coated with iron.

Cylinder Head Housing

The cylinder head housing **20** is secured to the upper end of the crankcase, as shown in FIG. **8**. The cylinder head housing **20** is bolted to the crankcase and provides a combustion chamber **201** above each cylinder **124**. A pair of exhaust valves **21** and a pair of intake valves **22** are mounted in each combustion chamber **201**. As shown in FIG. **11**, the pair of exhaust valves **21** are located on one side of the cylinder head housing **20** and the pair of intake valves **22** are located on an opposite side of the cylinder head housing **20**. The present invention, however, is not limited to a pair of exhaust valves and a pair of intake valves; rather, a single exhaust valve and a single intake valve may be employed. Furthermore, more than two intake and exhaust valves may be provided. Furthermore, any combination of intake and exhaust valves is contemplated provided each cylinder includes more intake valves than exhaust valves.

As shown in FIG. **8**, the intake valves **22** and the exhaust valves **21** are disposed at an angle with respect to the vertical axis of the engine **1** or **2**. This reduces the height of the cylinder head housing **20**, which reduces the overall height of the engine **1** or **2**.

The cylinder head housing **20** further includes at least one exhaust passageway **23** for each combustion chamber **201** extending through the cylinder head housing **20**, as shown in FIGS. **8**, **12** and **13**. The passageway **23** includes a pair of siamesed exhaust ports **231** that connect the exhaust passageway **23** to the chamber **201**, as shown in FIGS. **12** and **13**. Each of the pair of exhaust valves **21** is positioned in one of the respective exhaust ports **231** to selectively open and close the ports **231** at predetermined intervals to permit the removal of exhaust gases from the chamber **201**. An opposite end of the exhaust passageway **23** has an opening **232**, as shown in FIG. **14**, that is operatively connected to the exhaust manifold **30**. The exhaust manifold **30** is secured to the cylinder head housing **20** using suitable fasteners on a downwardly facing side of the cylinder head housing **20**, as shown FIG. **5**.

The cylinder head housing **20** further includes at least one intake passageway **24** for each cylinder **124** extending through the cylinder head housing **20**, as shown in FIGS. **8**, **12** and **13**. The passageway **24** includes a pair of siamesed intake ports **241** that connect the intake passageway **24** to the chamber **201**. Each of the pair of intake valves **22** is positioned in one of the intake ports **241** to selectively open and close the openings **241** at predetermined intervals to permit the influx of fuel and air into the chamber **201**. An opposite end of the intake passageway **24** has an opening **242**, as shown in FIG. **14**, that is operatively connected to the air intake and fuel injection system **40**. The air intake and fuel injection system **40** is secured to the cylinder head housing **20** opposite the exhaust manifold **30** using suitable fasteners on an upwardly facing side of the cylinder head housing **20**, as shown in FIG. **5**. While the intake and exhaust ports are shown as being siamesed, they can alternatively remain separated until connected to the respective intake and exhaust manifolds. The cylinder head housing **20** includes a spark plug assembly **28** that is located in a central inclined position, as described in greater detail below.

Valve Operating Assembly

A valve operating assembly illustrated in FIGS. **8** and **12–17** operates the intake valves **22** and exhaust valves **21** in accordance with predetermined engine operating parameters. The valve operating assembly is located within the cylinder head housing **20** and is driven by the crankshaft **123**. As discussed in greater detail below in connection with the power take off assembly **50**, the crankshaft **123** extends from the crankcase **10** into a power take off housing **59**. A gear assembly **54** is secured to the crankshaft **123** within the power take off housing **59** and includes a chain gear **542**.

A cam shaft **29** is rotatably mounted within the cylinder head housing **20**. One end of the cam shaft **29** extends into a control chain chamber **202** within the cylinder head housing **20**. The control chain chamber **202** extends into the cylinder block of the upper crankcase and enters the power take off assembly **50**. A cam gear **293** is operatively coupled to a chain gear **542** by a control chain **55**, which extends around both the gear **293** and gear **542**. The control chain **55** extends through the control chain chamber **202** into the power take off assembly **50**. The cam gear **293** and chain gear **542** are sized to have a 2 to 1 relationship.

The camshaft **29** is rotatably mounted to the cylinder head housing **20** in a position between the intake and exhaust valves **21** and **22**. Suitable bearing assemblies are provided for the smooth operation and rotation of the camshaft **29** within the cylinder head housing **20**. As shown in FIG. **12**, a plurality of cam lobes **291** and **292** are provided along the camshaft **29** to operate the valves **21** and **22** in each cylinder. A cam lobe **291** provides the necessary motion to operate the intake valves **22** through the rocker arm assembly **25**. A pair of cams **292** provide the necessary motion to operate the exhaust valves **21** through the rocker arm assemblies **26**. A cam **291** and a pair of cams **292** are positioned over each cylinder, as shown in FIGS. **16** and **17**. The cams **291** and **292** are oriented on the camshaft **29** to produce a predetermined timing for opening and closing the valves **21** and **22**. The orientation of the cams **291** and **292** vary for each cylinder such that all cylinders do not operate at the same time, rather the cylinders operate in a predetermined sequence. While the camshaft **29** is illustrated with a solid construction, it is contemplated that the camshaft **29** may have a hollow construction. Furthermore, the camshaft may be forged, cast or assembled.

The valve operating assembly includes a Y-shaped intake rocker arm assembly **25** that operates both of the pair of intake valves **22**, as shown in FIG. **13**, in response to the cam lobe **291**. The valve operating assembly further includes a pair of exhaust rocker arm assemblies **26** that operate the pair of exhaust valves **21**, as shown in FIG. **13**, in response to cam lobes **292**. The intake rocker arm assembly **25** is a forked assembly rocker arm having a pair of valve operating arms **251** and **252**. One operating arm **251** operates one of the intake valves **22** and the other operating arm **252** operates the other intake valve **22**. The fork like shape of the rocker arm assembly **25** provides access to the spark plug assembly **27** positioned within the cylinder head housing **20**. The spark plug assembly **27** will be described in greater detail below. The fork like shape of the rocker arm assembly **25** reduces the overall width of the necessary assemblies to operate the valves for each cylinder.

In an effort to reduce the weight of the rocker arm assemblies **25** and **26**, the rocker arm assemblies **25** and **26** may be produced from an aluminum alloy (AlSi) by forging or casting. The present invention, however, is not limited to rocker arm assemblies formed from aluminum; rather, it is contemplated that other materials including but not limited to steel and alloys of the same may be cast or forged to form the rocker arm assemblies **25** and **26**.

The rocker arm assemblies **25** and **26** are rotatably mounted on a rocker arm support axle **28** in a position between the intake and exhaust valves **21** and **22**. The stationary support axle **28** is mounted to the cylinder head by a plurality of fastener assemblies **281**, as shown in FIGS. **16** and **17**. The fastener assemblies **281** may include screw type fasteners, pin fasteners or other similar fastener assemblies for securing the support axle **28** within the cylinder head housing **20** and preventing its rotation. The rocker arm support axle **28** is mounted to the cylinder head housing **20**. The axle **28** is laterally offset and vertically spaced from the camshaft **29**, as shown in FIGS. **12**, **14** and **18**. This arrangement results in a compact construction that reduces the overall height of the cylinder head housing **20**. It is contemplated that the axle **28** may be located on the vertical axis of the cylinder or adjacent to the same.

The camshaft **29** is operatively connected to the crankshaft **123**, as described below. The cam gear associated with the crankshaft gear are sized to have a 2 to 1 relationship. The angled intake and exhaust valves **21** and **22** provide an enlarged area within the cylinder head housing **20** between the valves in which to locate the cam shaft, axle and the rocker arm assemblies **25** and **26**. This also provides sufficient space to maintain the 2 to 1 relationship between the cam gear and the crankshaft gear without increasing the height of the cylinder head housing **20**.

The rocker arm assembly **25** will now be described in greater detail, reference being made to FIGS. **12** and **14**. As described above, the rocker arm assembly **25** has a pair of operating arms **251** and **252**. A free end of each of the pair of operating arms **251** and **252** is positioned over a respective intake valve **22** and includes an hydraulic adjuster **253** for contacting the intake valve **22**. The hydraulic adjuster **253** abuts the upper surface of the valve stem of the intake valve **22**. The hydraulic adjuster **253** is located within a cavity **2511** and **261** in the respective arm **251** and **252**. A passageways **2512** and **262** extend from the cavities **2511** and **262**, respectively, to the rocker arm support axle **28**. The passageways **2512** and **262** are hydraulically linked to the rocker arm supportaxle **28**. The rocker arm support axle **28** includes a central passageway through which a supply of hydraulic fluid (preferably lubricant from the lubricant system) or other suitable lubricant flows. The fluid passes from the central passageway through radial openings **282** to the passageways **2512** and **262**. The fluid flows through the passageways **2512** and **262** to the cavities **2511** and **261** where it biases the hydraulic adjuster **253** into contact with the intake valve **22**. The fluid insures that the hydraulic adjuster **253** is always in contact with the intake valve **22** such that zero lash exists between the valve and hydraulic adjuster **253**. This insures that the entire motion of the cam **291** is transferred to the intake valves **22** to facilitate their opening and closing. Although fluid is used to bias the hydraulic adjuster **253** into engagement with the valves **22** in the embodiment illustrated, it is contemplated that a screw adjuster assembly or other mechanical assembly can be provided to perform the same operation.

An opposite end of the rocker arm assembly **25** includes a cam follower **254**. The follower **254** may include a roller assembly having bearings that is rotatably mounted to the rocker arm assembly **25**. The follower **254** travels along the cam **291**, which causes the rocker arm assembly **25** to pivot about the rocker support axle **28**. The motion of the cam **291** is transferred to open and close the intake valves **22**. Fluid from the central passageway **281** may be directed through another passageway, not shown, in the rocker arm assembly **25** to provide a supply of fluid to lubricate the follower assembly **254** to provide for smooth operation. The present invention, however, is not limited to the roller followers described herein; rather, it is contemplated that other followers including but not limited to sliding blocks may be utilized to follow the cam **291**.

The rocker arm assembly **25** has a compact angled construction, as shown in FIG. **14** so as to allow for a narrow and low construction. Similarly, the low arrangement of the camshaft **29** and associated drive chain wheel, which also does not project beyond the cylinder head housing **20**, as seen in FIGS. **16** and **17** assists in constructing an engine with a narrow and low profile.

As seen in FIGS. **8**, **12** and **14**, the camshaft **29** and the support axle **28** are offset relative to the longitudinal axis of the cylinder. The camshaft **29** is offset to provide room for the spark plug assembly **27**, described below. Both the camshaft **29** and the support axle **28** are located closer to the exhaust valves **21** than the intake valves **22**. The offset nature of the support axle **28** increases the overall length of the intake rocker arm assembly **25**. This increases the lever arm of the intake rocker arm assembly **25** and maximizes the force (within the size constraints of the cylinder head housing **20**) applied to operate both intake valves **22** with one rocker arm assembly. The intake and exhaust valves are disposed at an angle with respect to the cylinder axis. In principle, however, also other geometries (e.g. with a central arrangement of the camshaft **29**) are conceivable. Alternatively, the rocker arm support axle **28** may be located closer towards the intake valves so as to make the forked operating arms **251** and **252**—which are heavy due to this construction—shorter and thus less heavy. With this arrangement, the location of the camshaft **29** should also be relocated to maintain the lever arm of the intake rocker arm assembly **25**.

The rocker arm assemblies **26** will now be described in greater detail. Each exhaust rocker arm assembly **26** has the same construction. A free end of the rocker assembly **26** is positioned over a respective exhaust valve **21** and includes a hydraulic adjuster **263** for contacting the exhaust valve **21**. The hydraulic adjuster abuts the upper surface of the valve stem of the exhaust valve **21**. Like the hydraulic adjuster **253**, the hydraulic adjuster **263** is located within a cavity **261**. A passageway **262** extends from the cavity **261** to the rocker arm support axle **28**. The passageway **262** is hydraulically linked to the rocker arm support axle **28** through radial openings **282**. The fluid flows through the passageway **262** to the cavity **261** where it biases the operating assembly **263** into contact with the exhaust valve **21**. The fluid ensures that the hydraulic adjuster **263** is always in contact with the exhaust valve **21** such that zero lash exists between the valve and hydraulic adjuster **263**. This insures that all motion of the cam **292** is transferred to the exhaust valve **21** to facilitate opening and closing. Although fluid is used to bias the hydraulic adjuster **263** into engagement with the valve **21**, it is contemplated that a mechanical assembly (e.g. a screw adjuster) may be provided to perform the same operation.

An opposite end of the exhaust rocker arm assembly **26** includes a cam follower **264**. The follower **264** has a similar construction to the follower assembly **254**, described above. The rocker arm assembly **26** also has a compact angled construction, as shown in FIG. **14** so as to allow for a narrow and low construction.

The construction of the hydraulic adjusters **253** and **263** will now be described in greater detail in connection with FIG. **15**. The hydraulic adjusters **253** and **263** have the same construction. The hydraulic valve adjusters **253** and **263** are maintenance free and require no adjustment. The hydraulic adjuster **263** is positioned within the cavity **261**. The hydraulic adjuster **263** includes an inner stationary piston **2631** and an outer movable piston **2632**, which is located between the cavity **261** and the inner stationary piston **2631**. The inner stationary piston **2631** includes a central cavity **2633** that is in communication with the cavity **261**, as shown in FIG. **15**.

An opposite end of the piston **2631** includes an aperture **2634** such that the cavity **2633** is in fluidic communication with a cavity **2635** in the piston **2632**. A ball and seat check valve **2636** selectively closes the aperture **2634**. A valve contacting cap **2637** is pivotably mounted on an end of the piston **2632**. The cap **2637** contacts the valve stem of the exhaust valve **22** when the piston **2632** is in an extended position, as shown in the right side of FIG. **15**.

In operation, hydraulic fluid flows through channel **262** into the cavity **261**. After the cavities **261** and **2633** have filled with fluid, the valve **2636** opens to permit the flow of fluid into cavity **2635** through aperture **2634**. As the cavity **2635** fills with hydraulic fluid, the piston **2632** extends to the position shown in the right side of FIG. **15**. The spring assembly **2638** is located in the cavity **2635**. The downward travel of the piston **2632** is limited by contact with the valve stem and a seal **2639** that is secured to one end of the piston **2632** and is slidably received around the piston **2631**. When in the normal downward steady state position, the contacting cap **2637** contacts the valve stem such that motion of the rocker arm assembly is transferred to the valve to open the valve at predetermined locations of the camshaft **29**. After engine shut off, a sufficient amount of fluid is maintained in the cavity **2635** to maintain the outer movable piston **2632** in engagement with the corresponding valve stem.

FIGS. **16** and **17** illustrate an axial section through the camshaft **29** and the rocker arm support axle **28**. The camshaft **29** is mounted in a bearing bracket **293** with two collars **294** and **295**. Lubricant is supplied to the clearance region between these two collars **294** and **295**. By means of this double plain bearing in the respective bearing bracket **293**, the bearing becomes very rigid and the dynamic changing loads occurring during operation can be accommodated efficiently. Mounting of the camshaft **29** is effected by inserting it in from one end of the cylinder head housing **20** near the power take off end of the engine. The camshaft **29** is secured by a plate positioned within the cylinder head housing **20** against axial shifting. The plate extends through a vertical slot located within the cylinder head housing **20**. The plate may be further used to orient the axle **28** within the cylinder head housing **20**. It is also contemplated that a pin may be used to secure the camshaft against axial shifting. The pin may be located in a slot or groove extending around the perimeter of the camshaft.

Although the operation of the intake valves **22** and exhaust valves **21** has been described in connection with rocker arm assemblies **25** and **26**, other assemblies are contemplated for operating the valves. For example, the valves may be electromagnetically operated. Alternatively, the valves may be hydraulically operated using a slave piston/master piston arrangement. Furthermore, the Y-shaped rocker may be used to actuate the exhaust valves. Individual rocker arms may be used to operate intake valves. With this arrangement, the location of the spark plug assembly **27** must be relocated. It is also contemplated that gas springs may be used to bias the valves into a closed position when high rotation speeds are desired for high rpm output. It is also contemplated that a variable valve train may be substituted to vary the timing of the valve operation.

Spark Plug Assembly

The spark plug assembly **27** will now be described in greater detail in connection with FIG. **18**. A spark plug **271** is connected by threaded engagement to the cylinder head housing **20**, as shown in FIG. **18** such that an electrode portion of the spark plug **271** extends into the cylinder. The spark plug assembly **27** is located between the intake valves **22** and the exhaust valves **21** closer to the intake valves **21** because the intake side of the engine is cooler than the exhaust side of the engine. It is desirable to isolate the spark plug **271** from the remainder of the cylinder head housing **20**, which contains oil therein. A tube assembly **272** surrounds the spark plug **271**. The tube assembly **272** is preferably formed from a die cast plastic. It, however, is contemplated that other light weight materials may be used to form the tube assembly **272** so long as the tube assembly **272** isolates the spark plug **271** from the oil-carrying portions of the cylinder head housing **20**. It is preferable that the spark plug assembly **27** be inclined at an angle with respect to the central axis of the cylinder. The angle between the spark plug assembly and the intake valves is small (e.g. 3° is preferable). The angle, however, may be zero.

Each tube assembly **272** is sealingly inserted into a pedestal **273** on the cylinder head housing **20**, which forms a socket for the spark plug **271**. A slight compression fit between the tube **272** and a bore in the pedestal **273** can provide a sealing engagement between the two components although this sealing engagement can also be augmented by providing an o-ring between the two compartments. On an outer end, a seal **274** is vulcanized onto the tube assembly **272** which effects the sealing between the tube assembly **272** and a cylinder head cover **275**. Alternatively, the seal **274** can be provided as a separate component between the tube **272** and cover **275**. Use of the tube **272** provides for a lighter weight head assembly and also simplifies the casting of the cylinder head since the isolating tube is not cast as part of the cylinder head. The tube assembly **272** accommodates a plastic body spark plug connector **276** in which the ignition coil or the spark transformer are cast. In this way, the path of the high voltage to the spark plug **271** can be kept extremely short. From the outside, only a low voltage is supplied to the plastic body spark plug connector **276** and the ignition coil contained therein. The plastic body spark plug connector **276** and the spark plug **271** can easily be removed through the tube assembly **272**. The plastic body spark plug connector **276** abuts the inner side of the tube assembly **272**. A venting assembly is provided to enable venting from the spark plug region towards the environment. A splash water screen **2763** is attached to the plastic body **276**.

A cylinder head cover **275** is attached to the cylinder head housing **20** using a plurality of fastener elements **2571**, as shown in FIG. **19**. The cylinder head cover **275** is preferably formed from aluminum or some synthetic material. The connection between the cylinder head housing **20** and the cylinder head cover **275** is acoustically decoupled. An elastomeric gasket **2753** is positioned between the cylinder head housing **20** and the cylinder head cover **275** to provide a seal between the two components. The gasket **2753** has a protruding portion **2754** that is configured to sealingly engage a slot **2755** in the cylinder head cover **275**. This engagement maintains the gasket in a desired position between the cylinder head housing **20** and the cylinder head cover **275** and helps prevent the gasket **2753** from dislocating and causing leaks. In addition, the elastomeric gasket also reduces and prevents a direct sound propagation from the cylinder head housing **20** to the cylinder head cover **275** thereby reducing overall noise emanating from the engine. A further elastomeric gasket **2752** is provided between the fastener element **2751** and cylinder head cover **275** to seal the connection therebetween and also block direct sound propagation from the cylinder head housing **20** to the cylinder head cover **275** through the fastener **2751**. With this arrangement, the cylinder head cover **225** is isolated from the cylinder head housing **20**.

Exhaust Manifold

A preferred embodiment of the exhaust manifold **30** will now be described in connection with FIGS. **21–24**. The exhaust manifold **30** includes a first manifold **31** and a second manifold **32**, as shown in FIG. **24**. The first manifold **31** is connected to one side of the cylinder head housing **20**. It is preferably located on the smaller downward facing side of the cylinder head housing **20** because it does not require as much space as the induction system **40**, described below. The first manifold **31** includes at least one exhaust passageway **311** that is operatively coupled to each exhaust passageway **23** in the cylinder head housing **20**. Each exhaust passageway **311** connects to a main exhaust passageway **312**, which extends in a direction towards the power take off assembly **50**. With this arrangement, exhaust gases exit the cylinder head housing **20** through each exhaust passageway **23** when the respective exhaust valves **21** are opened. The exhaust gases then travel through the exhaust passageway **311** to the main exhaust passageway **312**.

The first manifold **31** is connected at the end nearest the power take off assembly **50** to the second manifold **32**. The second manifold **32** includes a main exhaust passageway **321**. The exhaust gases travel through the main exhaust passageway **321** into the muffler system **33**.

Due to US Government regulation, it is necessary to cool the exhaust components to limit the temperature of these components below a threshold value. It is desirable to cool the exhaust gases as the gases pass through the exhaust manifold **30** and an associated muffler system **33**. The muffler system **33** preferably includes a first muffler **331** directly connected to the exhaust manifold **30** and a second muffler **332** connected to the first muffler **331**.

The first and second manifolds **31** and **32** are equipped with an open loop cooling system for cooling the manifolds **31** and **32** and the exhaust gases contained therein. Each manifold **31** and **32** has a double jacket construction that permits cooling water to flow around the interior of the manifolds **31** and **32** without mixing with the exhaust gases. The first manifold **31** is preferably cast. The second manifold **32** is preferably formed from stainless steel.

The first manifold **31** has an inner manifold **313** and an outer manifold **314**, as shown in FIGS. **22** and **23**. The spacing between the inner and outer manifolds **312** and **314** forms a cooling passageway **315**. The inner and outer manifolds **313** and **314** are interconnected at various points along the manifold. The cooling passageway **315** has a generally u-shaped configuration when viewed from a vertical cross section such that it surrounds the main passageway **311** on the top, bottom and at least one side. The cooling water enters the passageway **315** through at least one inlet **316**. The cooling water then travels through the cooling passageway **315** and exits through at least one outlet **317**.

The second manifold **32**, as shown in FIG. **24**, also has an inner manifold **322** and an outer manifold **323**. The spacing between the inner and outer manifolds **322** and **323** forms a cooling passageway **324**, therebetween. The cooling passageway **324** substantially surrounds the main exhaust passageway **321**. The cooling water enters the cooling passageway **324** through at least one inlet **325** located near the connection between the first manifold **31** and the second manifold **32**. The cooling water exits the cooling passageway through at least one outlet **326** located near the point where the second manifold **32** enters the first muffler **331**.

The cooling system for the exhaust manifold **30** and muffler system **33** is an open loop cooling system. Cooling water is supplied to the first and second manifolds **31** and **32** by a jet pump of the propulsion unit of the personal watercraft **5**, which draws cooling water from the body of water in which the personal watercraft **5** is operating. An open loop cooling system can be used for the exhaust manifold **30** because the geometry of the cooling jacket for the exhaust manifold **30** is relatively simple with larger passageways. There is less concern for the clogging of these passageways. On the contrary, the geometry of the cooling system for the cylinder head housing **20** and crankcase **10** is more complex with smaller passageways. There is a greater concern about clogging that may occur when using a coolant drawn from outside the watercraft **5**. As such, a closed loop cooling system is preferred for the cylinder head housing **20** and crankcase **10**.

The cooling passageways **315** and **324** sufficiently cool the manifolds **31** and **32**. The temperature of the exhaust gases, however, remains too high. It must be further cooled before venting to the atmosphere or released into the water. It is desirable to cool the exhaust gases as the exhaust gases enter the first muffler **331**. At least one injection nozzle **34** is located adjacent the end of the main exhaust passageway **323**, such that a stream of cooling water is injected into the exhaust stream as the exhaust stream enters the first muffler **331**. Although it is preferable that the at least one injection nozzle **34** be located within the muffler **331**, it is contemplated that the injection nozzles **34** may be located within the main exhaust passageway **323**.

It is possible for the personal watercraft **5** to overturn or rollover during operation. It is desirable to prevent the cooling water used to cool the exhaust gases from traveling within the main exhaust passageways **314** and **323** to the cylinder head housing **20**. The design of the second manifold **32** and the connection between the second manifold **32** and the first muffler **331** prevent the return of the cooling water to the cylinder head housing **20**.

The second manifold **32** terminates within the first muffler **331** at a central location. The outlet opening for the main exhaust passageway **323** is spaced from the top, bottom and side walls of the first muffler **331**. With this arrangement, cooling water that has accumulated within the first muffler **331** should not enter the main exhaust passageway **323** because the cooling water should travel along the sides of the first muffler **331** (spaced from the outlet) when rollover occurs.

In the event that some cooling water enters the main exhaust passageway **323**, the configuration of the second manifold **32** prevents passage of cooling water to the cylinder head housing **20**. The second manifold **32** contains a unshaped bend or gooseneck portion that traps the cooling water. With this arrangement in a rollover condition, the cooling water must first travel downward from the first muffler **331** through the bend or gooseneck portion and then upward before entering the first manifold **31**. The change in direction of the main exhaust passageway **323** in the gooseneck portion essentially prevents any cooling water from entering the first manifold **31** or the cylinder head **32**.

The present invention is not limited to the above-described gooseneck portion for preventing water from entering the first manifold **31** at the cylinder head **20**; rather, other geometries that produce a similar effect are considered to be well within the scope of the present invention.

An alternative embodiment of the exhaust manifold will now be described in connection with FIGS. **43** and **44**. The exhaust manifold **300** is connected to one side of the cylinder head housing **20**. Like the manifold **30** described above, the manifold **300** is preferably located on the smaller downward facing side of the cylinder head housing **20**. The exhaust manifold **300** includes at least one exhaust passageway **310** that is operatively coupled to each exhaust passageway **23** in the cylinder head housing **20**. Each exhaust passageway **310** connects to a main exhaust passageway **320**. The exhaust gases exit the cylinder head housing **20** through each exhaust passageway **23** when the respective exhaust valves **21** are opened. The exhaust gases then travel through the exhaust passageway **310** to the main exhaust passageway **320**. The main exhaust passageway **320** first directs the exhaust gases toward the front of the personal watercraft, then in an opposite direction through knee bend **330** toward the rear of the personal watercraft. The exhaust gases may then exit the exhaust manifold **300** to a muffler system and/or water trap. The muffler system may include a pair of mufflers.

In this alternative arrangement, the exhaust manifold **300** also has a double jacket construction that permits cooling water to flow around the exhaust gases without mixing the cooling water and the exhaust gases. The double jacket construction includes an inner manifold **340** and an outer manifold **350**, which create a cooling chamber **370** therebetween. Webs **360** separate the cooling chamber **370** into a first portion **3701** and a second portion **3702**, as shown in FIG. **22**. The cooling water passes through the cooling chambers **3701** and **3702**, as shown in FIG. **44**.

Like the manifold **30** the exhaust manifold cooling system is an open loop cooling system. As such, a jet pump of the propulsion unit draws cooling water from the body of water in which the personal watercraft **5** is operating, shown in FIG. **44**. The cooling water is supplied to the exhaust manifold **300** through a primary inlet port **3510** located in the bend **330** of the exhaust manifold **300**, as shown in FIG. **43**. The cooling water then flows through the first chamber portion **3701** until it connects with the second chamber **3702** at the rear portion of the exhaust manifold **300**. The cooling water then flows back through the second chamber **3702** until it is discharged through the outlet port **3520** back into the body of water. Thus, the separation of the chamber **370** into two portions **3701** and **3702** that are interconnected only at an end of the exhaust manifold distant from the cooling intake and outlet ports provides for a U-shaped cooling circuit in the manifold, enhancing the cooling efficiency of the manifold.

These cooling arrangements maintain the exhaust manifolds **30** and **300** at a lower temperature than the cylinder head housing **20** and the cylinder block **10**. As a result, the exhaust manifolds **30** and **300** function as a heat sink, withdrawing heat from the cylinder head housing **20** and the cylinder block **10**. This reduces the cooling requirements placed on the closed loop cooling system **80**, described below. The coolant in the exhaust manifold (e.g. the water drawn from the body of water) has a lower temperature than the coolant for the closed loop cooling system, described below.

At least one temperature sensor **39** is located in the muffler to measure the temperature of the exhaust gases. The exhaust manifold **300** is equipped with an injection cooling system, which supplies additional cooling water to the exhaust manifold. A first injection nozzle **381** sprays cooling water directly into the exhaust passageway **320** in a direction away from the cylinder head housing **20**. A second injection nozzle **383** sprays cooling water directly into the exhaust passageway **320** also in a direction away from the cylinder head housing **20**. The location of the nozzles in the knee of the exhaust manifold prevents the backward travel of the cooling water into the cylinder head. The combined open loop cooling system with the injection cooling system functions to cool both the exhaust manifold and the exhaust gases within the manifold.

Air Intake and Fuel Injection System

The air intake and fuel injection system or induction system **40** will now be described in connection with FIGS. **26-31**. The system **40** is connected to the cylinder head housing **20** opposite the exhaust manifold **30**. The air intake into the engine **1** or **2** is effected from within the hull of the personal watercraft **5** via an air box, not shown, but disclosed in US Provisional Patent Application No. 60/224,355, filed on Aug. 11, 2000, entitled "WATERCRAFT HAING AIR/WATER SEPARATING DEVICE" and US Provisional Patent Application No. 60/229,340, filed on Sep. 1, 2000, entitled "PERSONAL WATERCRAFT HAING IMPROVED FUEL, LUBRICATION AND AIR INTAKE SYSTEMS" the specifications of which are incorporated specifically herein by reference. The air box comprises an air inlet in the form of a snorkel, a water separator unit and a muffler unit. The air box is located apart from the engine and connected to the engine via a tube or hose to prevent water from entering the air intake system.

The air flows through the tube connecting the air box with the engine, and then passes to an air intake manifold or plenum **41**, illustrated in FIGS. **26–31**. The air manifold **41** is preferably formed from a plastic material. The present invention, however, is not limited to the use of a plastic material; rather, metals, high strength alloys and other suitable synthetic materials may be used.

The air manifold **41** has a symmetrical geometry. With this arrangement, air flow into the air manifold **41** can be provided at either end of the air manifold **41**, thereby enabling use of the same air manifold **41** in either a normally aspirated engine **1** or a supercharged engine **2**, which engines have different flow paths for air into the air intake manifold. In the normally aspirated engine, the air from a throttle (if the engine has fuel injection) or a carburetor (if the engine does not have fuel injection) flows into one end of the air manifold **41**, as shown for example in FIG. **4**. Preferably, this end faces the airbox to shorten the distance and the pressure loss between the intake manifold and the airbox.

Irrespective of which end of the air manifold is used to intake air, in a fuel injection version of the engine, the air manifold **41** includes a throttle body **411** containing a throttle at the plenum inlet to regulate the flow of air into the manifold **41**. The degree of opening of the throttle of the throttle body **411** is controlled by the engine management system **200**. The throttle body **411** further includes a by-pass idle valve **4111**. The by-pass idle valve **4111** is preferably controlled by a stepper motor that controls the cross sectional opening of the by-pass idle valve **4111** and the amount of air flowing through it. Alternatively, it is contemplated that the idle valve **4111** may include an electromagnetically operated valve. The operation of the by-pass idle valve **4111** is controlled by the engine management system **200**. The engine management system operates the stepper motor based on the engine speed to adjust it to a given threshold value. In normal operation, the idle valve **4111** is open when the throttle of the throttle body **411** is closed. This permits the flow of a predetermined amount of air into the manifold **41** during an engine idling less than the normal air intake into the air manifold **41**. The idle valve **4111** is not fully closed when the throttle of the throttle body **411** is open. In a normal full load steady state operating condition, the idle valve **4111** is partly but not entirely open. This provides a reserve of intake air used for transient engine operating conditions (e.g., a rapid deceleration phase). The stepper motor is operated such that the maximum amount of air can be drawn into the air manifold **41** so that the air/fuel mixture is not too high. The location of the throttle body **411** is different for the normally aspirated engine **1** and the supercharged engine **2**. It is contemplated that the throttle body **411** may be replaced by a carbureter in a non-fuel injected version of the engine.

The air manifold **41** further includes at least one swing pipe **412** for each cylinder. Each swing pipe **412** is operatively connected to the respective intake passageway **24** to supply air to the combustion chambers through intake openings **241**. The flow pattern of the air within the air manifold **41** is indicated by the arrows in FIGS. **27–29** and **31**. As shown, the air enters the air manifold **41** via the throttle body **411**. The air passes radially through a cylindrical flame arrester **42** and then flows through each swing pipe **412** to the respective intake passageway **24**. The end cap **413** made be integrally formed with the air manifold.

The flame arrester **42** in the air manifold **41** prevents backfire of flames from entering the engine compartment interior within the hull of the personal watercraft. The flame arrester **42** includes a perforated inner pipe **421** and a pleated porous outer shell **422**. In accordance with the present invention, the location of the flame arrester **42** is advantageous. The flame arrester **42** is located within the central passageway in the air manifold **41**. As such, the flame arrester **42** is located between the swing pipe **412** and the air inlet. In the event of a backfire, this location is advantageous because all flames are caught by the flame arrester **42** before passage to the air inlet (i.e., the throttle or the supercharger). Thus, backfire flame cannot reach outside of the engine, especially important when the engine is installed on a watercraft or aircraft where an engine compartment fire can be more disastrous than in an automobile. Although a cylindrical flame arrester **42** is illustrated, it is also contemplated that the flame arrester may be in the form of a flat plate or an arcuate member.

The air manifold **41** is constructed to withstand the build up of back pressure in the event of a backfire. The manifold **41** is configured such that the back pressure is dissipated within the swing pipe **412**. To prevent failure or cracking of the manifold in the event of a significant build up of back pressure, a pressure relief valve may be provided. The pressure relief valve may be made integral with an end cap **413**, which is secured to an end of the air manifold **41**, as shown in FIG. **27**.

In the supercharger version of the engine **2**, the supercharger **90** and the throttle body **411** are interconnected between the air box and the air manifold **41**. The throttle body **411** is located between the air manifold **41** and the supercharger **90**. The supercharger assembly **90**, however, is connected to an opposite end of the air manifold **41**, as shown in FIGS. **30** and **31**. The location of the throttle body **411** is also relocated to this end. As such, the air manifold **41** is designed such that the throttle body **411** and the pressure relief valve, if provided, can be located on either end of the manifold **41** to provide increased flexibility such that the same manifold geometry can be used for either the supercharger version or the normally aspirated version of the engine.

The intake manifold **41** also includes at least one drainage port. The drainage plug is removably located within the drainage port. In the event that water enters the interior of the intake manifold **41**, the plugs can be removed to drain the water. Alternatively, a hose can be connected to the drainage port having a valve at an opposite end for more controlled drainage. Furthermore, it is contemplated that an automatically operated drainage valve may be provided to drain the air manifold upon engine shutdown.

It is contemplated that the air manifold **41** may include a cooling jacket **49** along an exterior wall of the air manifold **41**, as shown in FIG. **29**. The cooling jacket **49** cools the air within the air manifold **41** and, more particularly, the swing pipe **412** before entering the combustion chambers. The cooling of the intake air is especially useful for a supercharge version of the engine because the operation of the supercharger (by compressing) the air increases the temperature of the air. The cooling jacket may be linked to the open loop cooling system.

The air intake and fuel injection system **40** further includes a fuel injection assembly **43**. The fuel injection assembly **43** includes a common fuel rail **431**. The fuel rail **431** extends along an upper portion of the intake manifold **41**, as shown in FIGS. **26**, **27**, **30** and **31**. It is preferred that the pressure of the fuel into the fuel rail **431** be regulated by the fuel supply assembly **203** located in the fuel tank **204**. In an arrangement where the fuel supply is not controlled in the fuel tank, an optional pressure control valve **432** is located at one end of the fuel rail **431**. The pressure control valve **432** is provided to control fuel pressure within the fuel injection assembly **43**. In this arrangement, a separate fuel return line is required.

At least one fuel injection nozzle **434** extends from the fuel rail **431** to the each swing pipe **412** adjacent the connection to each intake passageway **24**. A fuel injection nozzle **434** is provided for each engine cylinder. The swing pipe **412** extends along the sides of the fuel injection nozzle **434**. This increases air flow around the injection nozzle **434** such that no pockets of reduced air flow are produced adjacent the nozzle **434** because reduced air flow may produce residue on the wall of the swing pipe adjacent the nozzle, which could reduce performance and flow of fuel into the cylinder chamber. Additionally, to prevent the formation of pockets, the nozzles **434** may extend into the swing pipe **412**. Fuel from the injection nozzle **434** is mixed with the air within the swing pipe **412** as the air enters the intake passageway **24**. The fuel injection nozzles **434** are electromagnetically controlled by the engine management system **200** so that the nozzles **434** are independently and sequentially operated.

Power Take Off Assembly

The power take off assembly So of the engine **1** or **2** will now be described in connection with FIGS. **32–34** and **36**. The crankshaft **123**, described above, extends from one end of the crankcase **10**, as shown in FIG. **33**. The rotation motion of the crankshaft **123** is transferred to a drive shaft **51**. A threaded connecting assembly **52** is secured to the end of the crankshaft **123**. The threaded connecting assembly **52** includes a plurality of teeth **521** that extend around an inner periphery of one end of the connecting assembly **52**. The teeth **521** are adapted to mate with complementary teeth **511** on the drive shaft **51**. As shown in FIGS. **36** and **37**, the teeth **511** have a generally arcuate shape. Although a generally linear tooth arrangement is considered to be well within the scope of the present invention, the arcuate tooth is preferred. The arcuate arrangement allows for slight angular deviations between the crankshaft **123** and the drive shaft **51**. This is especially important when the crankshaft **123** and the drive shaft **1** are not in exact alignment or when the personal watercraft is operated in extreme conditions, such as, for example, when jumping waves. The use of the threaded connecting assembly **52** is also advantageous. In the event of wear resulting from non-exact alignment, only the connecting assembly **52** need be replaced.

The arcuate teeth **511** of the connecting assembly **52** are lubricated with engine oil. The oil is supplied from a first crankshaft main bearing **1232** via hollow bores **1233** in the crankshaft **123**. The oil then flows to the arcuate teeth **511**. This arrangement reduces engine maintenance because the operator no longer needs to grease the connection between the crankshaft and the drive shaft. The lubrication is performed by the lubrication system of the engine. The power take off housing **59** seals the components contained therein with the power take off assembly **50**. Thus, protecting these components from exposure to marine conditions.

The connecting assembly **52** includes a sealing extension **522**, wherein the extension **522** extends along a portion of the drive shaft **51**. An o-ring seal **523** or other suitable sealing member is positioned between the sealing extension **522** of the connecting assembly **52** and the drive shaft **51**. There is no relative rotational movement between the drive shaft **51** and the connecting assembly **52**. As such, there are no rotational stresses on the o-ring seal **523**. The sealing extension **522** and the o-ring **523** prevents lubricant from escaping from the engine. A labyrinth sealing arrangement may be provided between the sealing extension **522** and the power take off housing **59** to prevent the passage of lubricant from the power take off assembly **50** around the drive shaft **51**. Alternatively, a screw or worm conveyor may be provided, which conveys lubricant back to the power take off assembly. At least one bore may be provided to form a shortcut such that the oil is drawn into the screw conveyor.

Additionally, the sealing of the drive shaft **51** with respect to the outside is effected by a sealing assembly **53**. The sealing assembly **53** includes several sealing elements that can be used alone or in combination. The sealing assembly **53** includes flexible bellows **531**, a shaft seal ring **532**, and sealing rings **533**. The flexible bellows **531** connects the power take off housing **59** with an external bearing carrier race **5311**, which in turn is rotatably mounted on the drive shaft **51** via two self lubricating antifriction bearings (rolling bearings) **5312** and a bearing carrier inner race **5313**. Sealing between the two bearing carrier races **5311** and **5313** is effected by the shaft sealing ring **532**. The sealing rings **533** (in the form of polymeric o-rings) act as a seal between the bearing carrier inner race **5313** and the drive shaft **51**. The sealing rings **533** also ensure a reliable fit between the two parts. A safety ring or clip **534** secures the bearing carrier inner race **5313** on the drive shaft **51** against any axial displacement. This may also be accomplished using a step formed in the drive shaft **51**. The flexible bellow **531** is clamped to the power take off housing **59** and the external bearing carrier race **5311** by clamps **5314** and **5315**, respectively.

Alternatively, the antifriction bearings **5312** are lubricated with engine oil. The oil is supplied from a first crankshaft main bearing **1232** via hollow bores **1233** in the crankshaft **123**. The oil flows through the arcuate teeth **511** to the antifriction bearings **5312** and finally returns between the power take off housing **59** and the connecting assembly **52** into the interior of the engine. With this arrangement, a second flexible seal is provided in the event the flexible bellow **531** fails.

The power take off assembly **50** further includes a gear assembly **54**, as shown in FIGS. **36** and **37**. The gear assembly **54** includes a main gear **541** secured to the crankshaft **123** for driving the balance shaft **115**, a chain gear **542** integrally connected to the main gear **541** for driving a cam control chain **55**, and a large gear **543**. It is contemplated that the chain gear **542** may be a separate component that is either force fit, fastened to or integrated into the crankshaft **123**. The large gear **543** includes at least a first gear **5432** for engagement with a starter **56** through intermediate gear **561**, as shown in FIG. **37** A second gear **5431** may be secured to the large gear **543** if the engine **2** is so equipped for driving a supercharger **90**, as described below and shown in FIG. **38** For reducing the number of required parts for the engine family, a single gear **543** having both gears **5431** and **5432** may be used in either the blown or normally aspirated engines. It is also contemplated that the large gear **543** is formed as a single gear such that a portion of each tooth of the gear is used to drive the supercharger and another portion is used to drive the starter.

Linking the intermediate gear **561** for the starter assembly **56** to the crankshaft **123** through the gear **543** results in a reduction of the engine profile. A thrust screw drive within the intermediate gear **561** for the starter assembly **56** allows for an automatic engagement of a drive pinion **562** with the first gear **5432** during the starting procedure. The intermediate gear **561** moves the drive pinion **562** into engagement with the first gear **5432** against the bias of a return spring **563**. At least one dampening spring **564** is provided to absorb vibration. After the starters operation is complete, the thrust screw drive disengages such that the return spring **563** biases the drive pinion **562** out of engagement with the first gear **5432**. The drive pinion **562** is mounted to a pinion shaft **565** that is connected to the starter assembly **56** such that rotational movement generated by the starter assembly **56** is transferred to the drive pinion **562**. The pinion shaft **565** is slidably and rotatably received within a recess in the power take off housing **59**.

As illustrated in FIG. **36**, a generator assembly **57** is also part of the power take off assembly **50**. The generator assembly **57** includes a magnet wheel **571** connected to the gear assembly **54**, as shown in FIG. **36** using suitable fasteners. The generator assembly **57** is a permanently excited 3-phase generator, in which permanent magnets **572**, which are fastened to magnet wheel **571**, rotate around a stator **573**. The stator **573** is fixed to the inner side of the power take off lid **59**. The location and arrangement of the generator assembly **57** provides for easy encapsulation because of reduced wiring requirements. The magnet wheel **571** rotates around the stationary coils. This arrangement is advantageous because it eliminates the need for rotating coil members and also in view of possible repair work. Furthermore, it reduces the weight of the rotating masses. Additionally, the magnet wheel **571** is constructed as an extrusion-molded part.

The rotational speed of the crankshaft **123** is measured by an engine or crankshaft speed sensor **58** located within the power take off housing **59**. A cup shaped actuator **544** is secured to the gear assembly **54** between the large gear **543** and the magnet wheel **571** of the generator assembly **57**. The actuator **544** extends between the gear **543** and wheel **571** and between the sensor **58** and the wheel **571**, as shown in FIG. **36**. The actuator **544** includes a plurality of teeth extending around the perimeter thereof. A predetermined number of teeth are missing at predetermined locations along the perimeter. The sensor **58** detects the absence of the teeth as the actuator **544** rotates. The speed of the crankshaft and engine speed can be determined from this.

Alternatively, it is contemplated that the magnet wheel **571** may include at least one conductor piece mounted therein. The conductor piece triggers the crankshaft or engine speed sensor **58**. Instantaneous values of the crankshaft position can be received therefrom and the angular speed (rotational speed) is then calculated by the engine management system **200**, described below. The angular resolution is 10°, i.e. during rotation of the crankshaft **123**, after every 10° of rotation, a pulse is sent by the crankshaft position sensor to the control device. It is contemplated that the present invention is not limited to an angular resolution of 10°; rather, angular resolutions greater than and less than 10° are considered to be well within the scope of the present invention.

The arrangement of the components within the power take off housing **59** results in a more compact engine design. As described above, the engine components are located on the power take off end. The power take off housing **59** protects these elements from the marine conditions in which the personal watercraft operates. Furthermore, a common drive assembly connected to the crankshaft **123** is provided to drive these components without the need for numerous belts and other connections. Additional features and benefits of the power take off assembly **50** will be described below in connection with the description of the lubricating system **60**, the blow-by ventilation system **70**, engine cooling system **80** and supercharger **90**.

Lubricating System

The lubricating system **60** will now be described in greater detail in connection with FIGS. **8**, **11**, **12**, **14–16** and **32–35**.

The engines **1** and **2** have a dry-sump lubricating system **60**. The lubrication system **60** includes the oil tank **11**, described above and shown in FIG. **8**. The oil collected in the crank chambers **121** emerges therefrom via outlet openings **111** into a channel **112**. The oil then flows to the upper portion **113** of the oil tank **11** adjacent the balance shaft **115**. From there, the oil flows back by gravity to the bottom of the oil tank **11**, where the oil is collected and stored.

From the oil tank **11**, the oil is conveyed to an oil cooling assembly **86**, shown in FIGS. **23** and **25**, by an oil pump **61**, as shown in FIGS. **25** and **33** through integrated channels in the lower crankcase **12**. The oil pump **61** is integrated into the power take off housing **59** and is coaxially disposed and driven by the balance shaft **115** via a connecting shaft **612**. The connecting shaft **612** is received within a suitable recess within the end of the balance shaft **115** such that rotation movement of the balance shaft **115** is transferred to the drive shaft **612**. The oil pump **61** is preferably a troichoid pump. It is preferred that the oil be sucked from the bottom of the oil tank **11**. Furthermore, it is also preferred that the oil be removed from a more centrally located pickup position within the tank **11**, rather than the front or rear of the tank **11**. This is a preventative measure to avoid air entrapment in extreme operating conditions (extreme acceleration and deceleration modes). The oil cooling assembly **86** is designed as a plate-type cooler and is fixed onto the cylinder block **10**. To cool the engine, water is used in a closed cooling system **80**, described in greater detail below.

From the oil cooling assembly **86**, the oil is conveyed to the oil filter unit **62**, as shown in FIGS. **32** and **34** through integrated channels in the lower crankcase **12**. The oil filter unit **62** has an oil filter casing **621** that is integrated to the power take off housing **59**. The oil filter unit **62** is closed at one end by a removable oil filter cover **622**. Located within the oil filter casing **621** is an annular oil filter **623** and a valve rod **624**. One end of the valve rod **624** is connected with the oil filter cover **622**. The valve rod **624** is secured to the cover by a suitable fastener. The valve rod **624** acts as a fastener to secure the cover **622** to the filter casing **621**. The other end of the valve rod **624** extends into a drainage opening **625**. When the valve rod **624** is pulled out of the drainage opening **625**, the oil which has remained in the filter casing **621** can automatically drain through the drainage opening **625**. Alternatively, the oil filter cover **622** may be configured as a screw lid.

Unlike conventional oil filter units where the overflow valve is integrated in the upper region of the filter cover **622**, the oil filter unit **62** includes an external overflow valve **626** and a bypass duct **627**. In the event that the oil filter unit **62** is clogged, a direct connection is formed between an inlet channel **628** and an outlet channel **629** of the oil filter unit **62**. This arrangement has the advantage that the oil does not flow around a dirty oil filter. Thus, no dirt particles can contaminate the oil circuit.

The filtered oil is then supplied to the engine **1** or **2** for lubricating the various components through the main oil gallery in the upper crankcase **13** of the crankcase **10**, as illustrated in the oil circuit in FIGS. **8** and **11**.

One aspect of the lubricating system **60** relates to the return of the oil from the crank chambers **121** in the upper crankcase **12** into the integrated oil tank **11**. The oil is pushed out of the crankcase. This is effected by a differential pressure acting between the crank chambers **121** and the oil tank **11** and the induction system, respectively. This differential pressure is a result of the pressure pulses caused by the pistons **1241** in the crank chambers **121**. It is also partially due to a consequence of a "Blow-By" effect, which refers to cylinder pressure losses. The piston **1241** does not provide a 100% sealing on the cylinder wall, so part of the combustion gas caused during combustion leaks past the cylinder downwardly into the lower crankcase **12**. This so-called blow-by gas creates additional pressure in the crank chambers **121** below the pistons **1241** and is dependent on the load and the rotational speed of the engine. However, on account of the above-mentioned blow-by effect, the overall effect results in a pressure that is always above the pressure between the air box and the throttle body. The return of the blow-by gas is described in greater detail below in connection with the blow-by ventilation system **70**.

The rotational movement of the crankshaft **123** is also utilized to carry oil to the outlet openings **111**, and here two effects are to be found. First, by the direct contact of the crank webs **1231** with the oil, in case of direct wetting, there occurs an entrainment effect as a consequence of the shearing forces. Second, with smaller amounts of oil in the crank chambers **121**, if there is no direct contact between crank web **1231** and oil, gas forces will occur which likewise drive the oil to the respective outlet openings **111**. At the base of the crank chambers **121**, in the vicinity of the outlet openings **111**, stripper edges may be arranged which strip the oil from the crank webs **1231**.

To enable an optimum utilization of the above-described effect for the oil return, the three crank chambers **121** (discussed above) in the crankcase **12** are hermetically separated from each other, and each crank chamber **121** is equipped with a separate outlet opening **111** for the oil. Thus, the pressure in one chamber is not affected by the pressure in the other chambers. The cross-sections of the channel system for the oil return following the outlet openings **111** are dimensioned suitably (i.e. not too large) so as to ensure the conveyance of the oil back to the oil tank **11** on account of the differential pressure, without the risk of a pressure equalization between oil tank **11** and crankcase **12**. Alternatively, the channels can also unify, so that one single channel **112** leads to the oil tank **11**. The arrangement should be designed such that no oil "shortcircuit" and no pressure balance will occur between the individual crank chambers **121**, i.e. oil must not be permitted to flow directly from one crank chamber **121** into another chamber.

The return channels **112** for the oil return from the three hermetically closed crank chambers **121** to the oil tank **11** may be realized by channels cast into the lower crankcase **12** which enter the oil tank **11** adjacent the union between the upper crankcase **13** and the lower crankcase **12**. Alternately, they may be realized by separate ducts, in particular hoses or tubes. As such, normally hoses are only used in connection with external oil tanks. In the present "in-case oil tank," hoses can be avoided. To prevent an undesired flow-back of oil from the oil tank **11** to the crank chambers **12** and—in consequence—a flooding of the crank chambers in extreme inclined positions or in flip-over position of the personal watercraft **5**, non-return valves (not illustrated) may be installed in the channels **112**.

To remove the lubricating oil which has collected in the region close to the bottom of the crank case **12** adjacent the bottom of the power take off housing **59**, a separate suction pump **71** is provided. Like the oil pump **61**, the suction pump **71** is coaxially arranged along and driven by the balance shaft **115**. The pump **71** is preferably a troichoid pump. The pump **71** is located on an opposite end of the balance shaft **115** when compared to the pump **61**. The oil is conveyed from the bottom of the power take off housing **59** through a duct **126** cast into the lower crankcase **12** to the suction pump **71**. Alternatively, it is contemplated that the blow-by gas created in the crank chamber **121** adjacent the power take off housing **59** is fed into the power take off housing **59** to provide pressure to remove the oil from the bottom of the power take off housing **59** near the bottom of the crank case.

The oil collected in the bottom of each crank chamber **121** exits through the opening **111**. The oil is then driven through the channel **112** back to the oil tank **11** by the blow-by gas pressure. The oil collected inside the power take off housing **59** is removed by a suction pump **71** or other suitable pumping assembly. The oil flows through a channel **126**, shown in FIGS. **11**, **41** and **49**, again integrated into the lower crankcase **12** from the power take off side to the opposite side, where the suction pump **71** is mounted, as shown in FIGS. **40** and **41**. The oil passes through an oil sieve **72** before it enters the suction pump **71** and is finally conveyed back through a U-shaped channel **711** to the oil tank **11**, as shown in FIGS. **11**, and **40**. It is contemplated that the channel **711** is integrated in the housing of the suction pump **71**.

Regarding the oil circuit, it is added that cooling and lubrication of the pistons **1241** and liners are effected by aid of spraying nozzles **64** at the lower side of the piston **1241**, as shown in FIG. **8**. Oil is supplied to the nozzles **64** from the main oil gallery **65**. The spray nozzle **64** is positioned such that the jet reaches the piston lower side not only in the lower dead center position illustrated, but also in the upper dead center position.

FIGS. **8** and **35** illustrate one possible oil channel system **63** in the region of the cylinder head housing **20** by way of a schematic 3D representation. Other systems are contemplated to be well within the scope of the present invention. The oil is conveyed to the cylinder head housing **20** through at least one ascending duct **631** in the upper crankcase **13**. The ascendind duct **631** is connected to the main oil gallery **65**. The oil enters cylinder head housing **20** from the ascending duct **631** through a transverse bore **632**. In the ascending duct **631**, a throttle **6311** is installed which restricts the amount of oil flowing therethrough. In addition, a check valve **6312** is disposed in the ascending duct **631**, which blocks the oil conduit as soon as the engine **1** or **2** is stopped. As such, a certain amount of oil can be stored in the channels in the cylinder head housing **20**. This stored oil is particularly useful during a cold start since lubrication can be initiated rapidly therewith and provided to the valve train sooner to prevent damage to the valve train.

Connecting bores **633** branch off of the transverse bore **632** and connect the latter with the bores **634**. The bores **634** also receive the cylinder head fastening screws. The oil rises upwardly in the annular gap between the cylinder head screw and the corresponding bores **634**. The oil then enters into a V-shaped channel section **635** formed by two obliquely downwardly directed bores **6351** and **6352**. From the ascending branch **6352** of the V-shaped channel section **635**, the oil directly enters into the interior of the hollow rocker arm support axle **28**. From there, the oil is directed to the bearing places of the roker arm assemblies **25** and **26** via the radial openings **282**, as shown in FIG. **14**. Also, the oil is admitted to the operating assemblies **253** and **263**. It is contemplated that other channel systems and arrangements are well within the scope of the present invention provided the channel systems conduct lubricant from the main oil gallery **65** to the support axle **28**.

Lubricant is supplied to the camshaft **29** via bearing bracket **293**, described above, through bore **636**.

Below the camshaft **29**, the oil ma y accumulate in a small basin in which the lobes **291** and **292** of the camshaft **29** may be immersed for lubricating purposes. The lubricant within the cylinder head housing **20** collects in a depression under the camshaft **29** adjacent the cylinder closest to the power take off assembly **50**. The oil from the other cylinders within the cylinder head flows to the depression through passageways **295**, which interconnect the areas in the cylinder head adjacent the other cylinders . The oil exits the cylinder head housing **20** through an inclined passageway into the control chain chamber **202** where it flows into the power take off assembly **50**. This lubricant contributes to the lubrication of the gears and supercharger **90**(if present) within the power take off assembly **50**.

Blow-By Ventilation System

The engines **1** and **2** are preferably equipped with a blow-by ventilation system **70** for separating oil from the vented blow-by gas. A preferred form of the blow-by ventilation system **70** is illustrated in FIGS. **3**, **4**, **11**, **40**, **41** and **46**.

The blow-by gas originating from the combustion chambers **124** due to leakage between the pistons **1241** and cylinder liners first accumulates in the (sealed) crank chambers **121** and from there it flows together with the oil through the channels **112** to the oil tank **11**, where it accumulates and mixes in the upper portion **113** of the oil tank **11** with any gas in the oil tank **11** from the power take off assembly **50**. From the oil tank **11**, the gas mixture is then conveyed through a channel **712** (in the housing of the suction pump **71** and the lid of the sieve **72**), shown in FIG. **40** to a shutoff and pressure relieve valve **73**, which is open in normal engine operation. The pressure relief valve **73** includes a valve rod **731** that moves the valve **73** between open and closed positions by a solenoid assembly **77**. In the event that the solenoid assembly **77** is not operational, the pressure relief valve **73** includes a spring assembly **732** that permits the opening of the valve **73** in the event of a build up of pressure within the tank **11**.

The gas mixture from the oil tank **11** is split into two partial flows: a first portion flows back to the cylinder head chamber within the cylinder head housing **20** through a passageway **74**, shown in FIGS. **40** and **41**. A second portion is vented tangentially into an oil separator **75** designed as a cyclone. In the cyclone, the gas mixture is separated from oil by centrifugal forces due to the swirling of the gas/oil mixture in the cyclone. The cleaned gas mixture leaves the cyclone through a central pipe **751**. The cleaned gas mixture then passes a second shutoff and pressure relief valve **76** and is finally conveyed to the air intake between the airbox and the throttle body **411**, where it merges with the fresh air drawn in by the engine.

The shutoff and pressure relief valve **76** is also mounted on the valve rod **731** and is also actuated by the solenoid **77**. With this arrangement, the valves **73** and **76** operate simultaneously. The valves **73** and **76** are closed by drawback springs **732** and **761** when the solenoid **77** is not activated and they are open when the solenoid **77** is activated. With this arrangement, the engine is sealed, preventing oil leaks when the engine is shut down. In normal (upright vehicle) engine operation, the solenoid **77** is activated and the valves **73** and **76** are opened respectively. However, in the event of a roll-over of the vehicle, the valves are closed instantly to prevent oil from entering the induction system **40** and/or the airbox and leaking into the environment. The closure of valve **73** prevents oil from accumulating in the cylinder head housing **20** in a roll-over event. This would cause a temporary lack of oil in the oil tank **11**, when the personal watercraft **5** has returned to a normal upright position and could result in an undersupply of lubricant to the engine, which may result in severe damage to the engine **1** or **2**. The valves **73** and **76** are also closed when the engine is shut down.

A pressure sensor or sensor switch may be provided in the oil tank **11** or in the channel **712** to sense the pressure within the tank **11**. If the oil pressure exceeds a certain threshold value, the engine management system **200** operates in an emergency mode (e.g. limp home function). The engine management system operates the engine at a reduced speed. The engine management system also interacts with other onboard computers systems to notify the operator of the engine malfunction. Additionally, the pressure sensor can be used to detect oil leakage in the lubrication circuit.

The gas mixture enters the upper portion of the cyclone **75** through the opening **755**. As such, the gas mixture tangentially enters the cyclone **75**. Oil droplets within the gas mixture are thrust against the inner wall of the cyclone **75** as a result of centrifugal forces within the cyclone **75**.

The separated oil then flows down the inner wall of the cyclone **75** towards opening **752**; collects in the bottom of the cyclone **75**; and exits the cyclone **75** through an opening **752** into a channel **753** integrated in the sieve lid **721**, and merges with the oil flow from the power take off assembly **50** in front of the oil sieve **72**, to be conveyed back to the oil tank **11**. Within the channel **753** there is provided a throttle **754** which ensures that a sufficient height negative pressure (vacuum) can build up in the suction port of the suction pump **71**, so that the power take off housing **50** is drained reliably in all operating conditions. In a cold start condition (when the oil is very viscous) the throttle **754** may even be closed by an additional valve (not shown) especially at idling speed to guarantee the aforesaid requirement.

An oil filler tube **78** is integrated to the cyclone **75**. A cap **781** is provided for closing the filler tube **78**. Fresh oil flows down the filler tube **78** into a channel **722** integrated in the sieve lid **721**. The oil enters a U-shaped duct through a port **715**, shown in FIG. **40**, in the housing of the suction pump, merges with the oil from the power take off assembly **50** and is finally conveyed to the oil tank **11**.

In the preferred embodiment, the valves **73** and **76**, the cyclone **75** and the oil filler tube **78** are assembled to form a single unit.

In accordance with the blow-by gas ventilation system **70** described herein, a slight vacuum (underpressure, negative pressure, subpressure) is generated in the interior in the power take off assembly **50** and within the cylinder head housing **20**. As a result, no oil or contaminated blow-by gas can escape to the environment.

Engine Cooling System

An engine cooling system **80** will now be described in connection with FIGS. **25**, **32** and **33**. The engine cooling system **80** is a closed system utilizing a coolant such as glycol, water or a mixture of them. The present invention, however, is not limited to these coolants; rather, it is contemplated that other cooling liquids are considered to be well within the scope of the present invention. The cooling circuit of the engine cooling system **80** is illustrated in FIG. **25**. The closed loop cooling system **80** cooperates with the open loop cooling arrangement described above in connection with the exhaust manifold **30** to effectively cool the engines **1** and **2**.

The engine cooling system **80** includes a pump assembly **81** located on one end of the engine **1** or **2**, as shown in FIG. **32**.

As illustrated in FIG. **33**, the pump assembly **81** is arranged externally of the power take off housing **59**. The power take off housing **59** and pump lid **611** together form the pump casing. It is designed as a rotary pump and consists of an impeller **811** which is located, screwed or attached onto the end of the connecting shaft **612**, which projects from the power take off housing **59**. The connecting rod **612** also drives the oil pump **61**. Impeller **811** is driven by connecting rod **612**. The connecting rod **612** also drives the oil pump **61**. The pump assembly **81** also includes a pump lid **812**, which is fastened to the power take off housing **59** and forms the pump casing in cooperation therewith. The pump assembly **81** has a one piece housing having an integrated thermostat.

As shown in FIG. **25**, the coolant flows from the pump assembly **81** through a passageway **82** to the cylinder block of the upper crankcase **13**. The passageway **82** includes a main passageway **821** and a by-pass passageway **822**. The passageways **821** and **822** direct coolant to the cooling passageway **125** in the cylinder block. The coolant flows along the exterior of the cylinders **124**, as shown in FIG. **25**. With this arrangement, the coolant travels in a generally U-shaped manner along a side of the cylinders **124** adjacent the intake manifold; around the end of the cylinder furthest from the power take off assembly **50** and then along the side of the cylinders adjacent the exhaust manifold in a direction back towards the power take off assembly **50**. At the same time, the coolant is directed in an upward direction towards the cylinder head housing **20**. The by-pass passageway **822** reduces the load on the main passageway **821** and improves the flow pattern in the cooling passageway **125** at an end portion of the cooling passageway **125** opposite the inlet. The coolant from the by-pass passageway **822** mixtures with the coolant in the coolant passageway **125** to reduce the temperature of the coolant in the end portion of the cooling passageway **125**. Furthermore, the entry of coolant into the cooling passageway **125** from the by-pass passageway **822** improves the upward flow of coolant into the cylinder head housing **20**. It is preferred that the passageways **821** and **822** are integrally formed in the power take off housing **59** and crankcase **10**. It, however, is contemplated that the passageways may be hoses connecting the components to one another.

From the upper crankcase **13**, the coolant then passes upwardly to the cylinder head housing **20** through bores **131** in a head gasket **130** positioned between the upper crankcase **13** and cylinder head housing **20**, as schematically illustrated in FIG. **25**. The bores **131** are located on the exhaust manifold side of the gasket **130**. These bores **130** act as throttles to adjust the flow of coolant into the cylinder head housing **20**. Additional small bores are located on the intake manifold side of the gasket **130**. These bores vent air trapped within the passageway **125** into the cylinder head housing **20**. The coolant first passes over the exhaust side of the cylinder head toward the intake side of the cylinder head before exiting the cylinder head housing **20** through a common passageway.

From the cylinder head housing **20**, the coolant is then conveyed through a hose to a thermostat **83** through an inlet passageway **817** located on the pump assembly **81**, as shown in FIGS. **25** and **32**. As illustrated in FIG. **33**, the thermostat **83** is directly mounted on the pump lid **812**. The thermostat **83** comprises a two-part thermostat casing **831** and **832** including hose connections and a temperature-sensitive valve **833**, which automatically opens if a predetermined temperature threshold value is exceeded. The coolant then flows through outlet passage **816** to a heat exchanger **84** (shown schematically in FIG. **25**), where the coolant is cooled by exchanging heat to the atmosphere. This can be in the form of a cooling plate exposed to the body of water. The cooling plate may be located in a lower portion of the hull of the personal watercraft **5**. The cooling plate is described in U.S. Provisional Patent Application Ser. No. 60/160,819, filed Oct. 21, 1999 entitled "WATERCRAFT WITH CLOSED-LOOP HEAT EXCHANGER," and U.S. patent application Ser. No. 09/691,129, filed Oct. 19, 2000 entitled "WATERCRAFT HAVING A CLOSED COOLANT CIRCULATING SYSTEM WITH A HEAT EXCHANGER THAT CONSTITUTES AN EXTERIOR SURFACE OF THE HULL" the specifications of which are incorporated herein specifically by reference. The coolant is then returned to the pump assembly **81** through an inlet **815**.

The primary purpose of the cooling system **80** is to cool the engine **1** or **2** during operation. The operation of the cooling system **80** is temporarily modified during engine start-up so that the engine quickly reaches an optimal operating temperature. During initial engine start-up, the thermostat **83** deactivates the heat exchanger **84**. As such, the coolant is not cooled prior to reentry into the pump assembly **81**; rather, the coolant returns directly from the inlet **817** into the coolant pump **81**.

The cooling system **80** furthermore includes an oil cooling assembly **86**. The oil cooling assembly **86** is connected to pump assembly **81** and thermostat **83**. With this arrangement, a portion of the coolant from the pump assembly **81** is directed to the oil cooling assembly **86** through passageway **861** to cool the engine oil. After passing through the oil cooling assembly **86**, the coolant returns to the thermostat **83** via return passageway **862**. The coolant from the passageway **862** enters the thermostat housing in the vicinity of the inlet **817**. The oil cooling assembly **86** preferably is a plate-type cooler and disposed on the side of the lower crankcase **12**. The coolant, which heats sooner than the oil, is used to heat the engine oil during engine start-up.

The cooling system **80** further includes a temperature sensor **87**, which is linked to the engine management system, shown in FIGS. **25** and **42**. As shown in FIG. **25**, an expansion reservoir **88** is provided in the return from the cylinder head housing **20** to the thermostat **83**, as shown in FIG. **23**. The expansion reservoir **88** adjusts for expansion of the cooling fluid within the system **80**. The expansion reservoir **88** further a refill port **881** for refilling the system **80**. The reservoir **88** further provides a venting function for removing air from the cooling system **80**. In this manner, the interconnecting duct between the reservoir **88** and the cylinder head housing **20** has to be linked to the highest point in the cylinder head housing **20** to prevent the formation of an air barrier which could cause overheating.

Supercharger Assembly

As discussed above, the engines in accordance with the present invention may include a supercharger **90**. The engine **2** having a supercharger **90** is illustrated in FIGS. **6**, **7**, **30**, **31** and **38**. The supercharger **90** is provided to increase the air intake and enhance engine performance. The preassembled supercharger **90** is plugged in a corresponding port **591**, as shown in FIG. **33**, in the power take off housing **59** and sealed with sealing rings **592**, as shown in FIG. **38**. It is contemplated that a turbocharger may be used in connection with the present invention. The supercharger, however, provides improved operating characteristics when compared to the turbocharger. Furthermore, the turbocharger produces additional heat as compared to the supercharger, which places increased demands on the cooling systems.

The supercharger **90** includes a cast housing **91**, which is preferably formed from a metal, however, it may be formed from a high strength plastic or other suitable material. The housing **91** includes an inlet portion **911**. The inlet portion **911** is operatively connected to the airbox (not shown). Air enters the supercharger **90** through the inlet portion **911**. Located within the housing **91** adjacent the inlet portion **911** is an impeller **92**, which operates to draw air into the supercharger from the airbox. An air passageway **912** extends around the impeller **92** to collect the air compressed by the impeller. The air passageway **912** is connected to the intake manifold **41** through the throttle body **411**. The housing **91** further includes a mounting portion **913** that extends backward from the inlet portion **911**. The mounting portion **913** is received within the port **591** in the power take off housing **59** and sealed with at least one sealing assembly **592**.

As shown in FIG. **38**, a blower drive shaft **922** extends through the mounting portion **913** and inlet portion **911**. The blower drive shaft **922** is rotatably mounted within the housing **91** with at least one bearing assembly **921**. A drive pinion **93** is coupled to the blower drive shaft **922**. It is preferred that this be a non-positive coupling. As such, the drive pinion **93** is non-positively connected with the blower shaft **922** via an intermediate element **94** by a biasing spring force, which is preferably supplied by a spring assembly **95**. The spring assembly **95** includes a plurality of cup springs. Other spring assemblies and means for providing a connection that can slip under high torque to prevent damage to the impeller or other components, however, are considered to be well within the scope of the present invention. The drive shaft **922** includes splines to prevent rotational movement of the intermediate element **94** with respect to the drive shaft **922**. The shaft **922** includes a lubrication passageway that delivers lubricant to the drive pinion **93** to reduce wear. The lubrication passageway is connected to the lubrication system. The connection between the drive pinion **93** and the intermediate element **94** is formed as a plane frictional surface. This unique connection assembly can dampen the rotational and torsional vibrations transmitted by the crankshaft **123**.

The supercharger **90** is operatively coupled by the drive pinion **93** to the gear assembly **54** through gear **5431**. The supercharger **90** preferably includes a cooling jacket connected to the open or closed loop cooling system to cool and prevent failure of the supercharger **90**. The cooling of the supercharger **90** improves engine performance.

In accordance with the present invention, the supercharger **90** preferably utilizes a low-cost rotary (radial or radial-axial) blower. The present invention, however, is not limited to these blowers; rather, it is contemplated that a positive displacement blower (e.g. a Rootes or Wankel blower) may be employed. Furthermore, the supercharger **90** may be used for separating a certain water content from the intake air.

Control Tensioner

In accordance with the present invention, the engines **1** and **2** are preferably equipped with a control tensioner for controlling the tension within chain **55**. The present invention, however, is not limited for use with a chain; rather, it is contemplated that the control tensioner can be used with other flexible linkages, including but not limited to belts. A mechanical chain tensioner **100** is illustrated in FIG. **39**. The tensioner **100** includes a driving element **101**. The driving element **101** preferably includes a spring assembly. The spring assembly is preferably a rotationally active helical pressure spring. The spring assembly **101** is rotationally biased by aid of a thread cap **102**. The spring includes a spring ender **1011** that engages a slot **1021** in thread cap **102**. The thread cap **102** is externally screwed into a retainer **103**. The spring assembly **101** is received at one end in a blind hole bore of a hollow adjustment element **104** which is screwed into a thread bore of the retainer **103**. The spring also includes a spring end **1012** that engages a slot **1042** in adjustment element **104**. The overlapping thread engagement of adjustment element **104** with retainer **103** is designed to be relatively long. As oil gets into this threaded connection, it provides a small damping effect to the adjustment element **104** due to vibrations of the cam chain. This small damping effect is enhanced if the thread overlap is kept relatively long. The external thread of the adjustment element **104** preferably includes multiple threads and it is designed such that it is borderline self-locking in the retainer **103**. This design must take into account the presence of oil between the threads, which reduces friction, when determining the necessary inclination of the threads. If the inclination is too small (very self locking), a strong spring force is required to overcome the locking action of the threads. It is desirable to avoid unnecessary tension on the chain to avoid wear and decreases in the lifetime of the chain. The self tensioning action is effected by the interaction of the chain vibration and the borderline self locking of the threads. That is, it will maintain its extended position under normal loads but can retract a distance under high loads to prevent damage to the cam chain. For instance, if automatic adjustment occurs when the engine is cold, upon reaching operation temperature, the aluminum cylinder and head have expanded more than the steel cam chain and can create too high of a tension in the chain. The borderline self locking feature allows the plunger to retract slightly before chain tension becomes so high as to damage the chain. The adjustment element **104** is rotationally driven by the spring assembly **101** if the tension of the chain **55** slackens and is axially outwardly displaced. The adjustment element **104** acts via a balancing arcuate intermediate piece **105** on a tensioning rail **106**. The chain tensioner **100** enables a later adjustment by aid of the combined biasing and fixing element **102** if the chain **55** undergoes elongation.

The thread piece **102**, the retainer **103** and adjustment element **104** preferably are made of synthetic material because of the smaller thermal elongation encountered as compared to aluminum. The adjustment element **104** includes a steel insert **1041** on one end to reduce wear.

In accordance with the present invention, the engines **1** and **2** described herein are not limited to the mechanical chain tensioner **100**; rather, other tensioner assemblies are contemplated to be well within the scope of the present invention. For example, a hydraulic tensioner may be used. The mechanical tensioner **100**, however, has numerous advantages over this hydraulic counterpart. First, the mechanical tensioner **100** can be manufactured at a lower cost and does not require a complicated oil supply.

Engine Control Unit

The operation of the engine **1** or **2** is controlled by an engine management system **200**, as shown in FIG. **42**. The engine management system **200** includes an electronic control unit **201** monitors and controls the operation of various engine components including but not limited to ignition, the fuel pump, the fuel injection assembly, the air intake, engine cooling, engine speed, engine lubrication, exhaust gas in the muffler in response to input from various sensors and monitors located with the engines **1** and **2**. It is contemplated that the electronic control unit **201** may further control functions, such as, e.g., realization of a departing lock, realization of a start/stop control, and the identification of authorized personal watercraft users. The electronic control unit **201** further communicates with the other computer systems on the personal watercraft for the control of instruments, non engine watercraft functions and service needs.

The engine management system **200** also controls the gas pump **203** in the gas tank **204**, which includes a coarse filter **2041** and a float assembly **2042**.

The gas pump **203** has an associated pressure regulator **2043**, such that a constant gas pressure is mechanically provided. From there, a returnless fuel system **205** leads to the injection nozzles or valves **434** seated on the fuel rail **431**. These injection nozzles **434** inject the fuel in the form of jets in the air in the intake passageway. The engine management system **200** controls the operation of the nozzles **434** such that there is sequential injection, wherein each cylinder has an individual injection (i.e., no group injection). The injection amount is determined by the engine control device **201** on the basis of the applied characteristic fields by the pulse width, i.e. by the duration of the injection time.

A returnless fuel system **205** prevents the fuel from heating due to the engine heat, as could otherwise be the case with a fuel return from the engine to the fuel tank.

The engine management system **200** also includes various sensors, such as the temperature sensor **39** in the exhaust muffler, an air temperature sensor **43** attached to the intake manifold **41** and a water temperature sensor **87**.

A knock sensor **206** senses at an early time the knocking critical for the engine—which has a high specific performance level. The knock sensor **206** includes a piezo quartz element, which measures the solid-borne acoustic signals at the cylinder block and transmits the corresponding signals to the electronic control unit **201**. The latter has a detection software to detect a possible knocking combustion and to cause a correction in a manner known per se, by ignition angle displacement.

The sensors further include the crankshaft position sensor **207**. A corresponding rotary position sensor **208** is associated with the camshaft. By aid of this camshaft sensor **208**, it is recognized whether the crankshaft is present in the angle range of 0 to 360° or in the range of 360 to 720°, which is possible via the camshaft because the latter rotates at half the rotational speed of the crankshaft. For the sake of simplicity, the camshaft sensor **208** is directly associated with the chain wheel **551** at the camshaft.

For load measurement, the actual load of the engine is calculated by the intake manifold pressure measured by sensor **210** and engine speed measured from the crankshaft **123** in the power take off assembly **50**. A throttle potentiometer **209** is used for corrections and a limp home function. In the event the engine is operating in a limp home function (e.g., broken intake air pressure sensor), the engine control unit **201** communicates with another onboard computer system to notify the operator via an instrument panel that the engine is operating in a limp home function. A pressure sensor **210** is arranged in the suction pipe to sense the absolute pressure, which is especially useful for the engine **2** containing the supercharger assembly **90** and for all operation modes with slightly opened or closed throttle valve. Thus, there is no direct air amount or air mass measurement, but auxiliary parameters are used therefor.

Finally, for the sake of completeness, various voltage checks should be mentioned which are carried out by the electronic control unit **201**, e.g. for the supply voltage of the injection valves, which is useful insofar as the board voltage on the personal watercraft **5** may very well fluctuate.

It will be apparent to those skilled in the art that various modifications and variations may be made without departing from the scope of the present invention. Thus, it is intended that the present invention covers the modifications and variations of the invention, provided they come within the scope of the appended claims and their equivalents.

What is claimed is:

1. A four stroke internal combustion engine, comprising:

a crankcase having a crank shaft rotatably mounted therein;

a cylinder head connected to the crankcase, wherein the crankcase and the cylinder head form at least one cylinder;

at least one intake valve for the at least one cylinder, wherein the at least one intake valve has an intake valve axis;

at least one exhaust valve for the at least one cylinder, wherein the at least one exhaust valve has an exhaust valve axis;

a valve actuation assembly located in the cylinder head for operating the at least one intake valve and the at least one exhaust valve, wherein the valve actuation assembly is located substantially between the intake valve axis and the exhaust valve axis, the valve actuation assembly comprising

a cam shaft rotatably mounted within the cylinder head,

a support axle mounted within the cylinder head, offset from the cam shaft;

at least one exhaust rocker arm pivotally mounted on the support axle,

at least one intake rocker arm pivotally mounted on the support axle,

wherein the cam shaft is operatively coupled to the crankshaft such that rotational movement of the crankshaft is transferred to the cam shaft,

wherein the at least one exhaust rocker arm is operatively coupled to the cam shaft for operating the at least one exhaust valve, and

wherein the at least one intake rocker arm is operatively coupled to the cam shaft for operating the at least one intake valve; and

a lubrication system for lubricating the engine,

wherein the lubrication system includes a supply of lubricant to the cylinder head,

wherein the support axle has a central passageway extending therethrough, and

wherein at least a portion of the supply of lubricant flows through the central passageway in the support axle.

2. The four stroke internal combustion engine according to claim **1**, wherein each cylinder has a longitudinal axis, wherein each of the at least one intake valve and the at least one exhaust valve is disposed at an angle with respect to the longitudinal axis.

3. The four stroke internal combustion engine according to claim **2**, wherein the engine includes a pair of exhaust valves for each cylinder and the valve actuation assembly includes an exhaust rocker arm for each exhaust valve.

4. The four stroke internal combustion engine according to claim **3**, wherein each exhaust rocker arm comprises:

a cam follower located on one end of the exhaust rocker arm,

wherein the cam follower is adapted to follow a profile of an exhaust cam lobe located on the camshaft; and

an exhaust hydraulic adjuster located on an opposite end of the exhaust rocker arm, wherein the exhaust hydraulic adjuster is adapted to contact and operate the exhaust valve in response to movement of the exhaust rocker arm by the exhaust cam lobe.

5. The four stroke internal combustion engine according to claim **2**, wherein the engine includes a pair of intake valves for each cylinder and the valve actuation assembly includes a forked intake rocker arm for the pair of intake valves.

6. The four stroke internal combustion engine according to claim **5**, wherein the forked intake rocker arm comprises:

a cam follower located on one end of the intake rocker arm, wherein the cam follower is adapted to follow a profile of an intake cam lobe located on the camshaft;

a first actuator arm for operating a first of the pair of intake valves;

a second actuator arm for operating a second of the pair of intake valves; and

a pair of intake hydraulic adjusters located on an opposite end of the forked intake rocker arm,

wherein each intake hydraulic adjuster is adapted to contact and operate the one of the pair of intake valves in response to movement of the intake rocker arm by the intake cam lobe.

7. The four stroke internal combustion engine according to claim **6**, wherein each of the intake hydraulic adjusters comprises:

an intake piston actuator slidably received within a cavity in an end of the actuator arm, and a fluid passageway extending through the actuator arm to the support axle, and

wherein the central passageway in the support axle is in fluidic communication with the fluid passageway such that fluid within the central passageway flows through the fluid passageway to the cavity to bias the intake piston actuator into engagement with the intake valve.

8. The four stroke internal combustion engine according to claim **7**, wherein the exhaust rocker arms are rotatably mounted on the support axle on opposite sides of the intake rocker arm.

9. The four stroke internal combustion engine according to claim **8**, wherein each exhaust rocker arm comprises:

a cam follower located on one end of the exhaust rocker arm,

wherein the cam follower is adapted to follow a profile of an exhaust cam lobe located on the camshaft; and

an exhaust hydraulic adjuster located on an opposite end of the exhaust rocker arm, wherein the exhaust hydraulic adjuster is adapted to contact and operate the exhaust valve in response to movement of the exhaust rocker arm by the exhaust cam lobe.

10. The four stroke internal combustion engine according to claim 6, wherein the first actuator arm is spaced from the second actuator arm.

11. The four stroke internal combustion engine according to claim 10, further comprising:

a spark plug assembly disposed in the cylinder head, wherein the spark plug assembly is positioned between the first actuator arm and the second actuator arm.

12. The four stroke internal combustion engine according to claim 11, wherein the longitudinal axis of the spark plug assembly extends substantially parallel to the longitudinal axis of the intake valves.

13. The four stroke internal combustion engine according to claim 11, wherein the longitudinal axis of the spark plug assembly and the intake valve axis form an acute angle not greater than 20°.

14. The four stroke internal combustion engine according to claim 11, wherein the spark plug assembly comprises:

a tube assembly secured to the cylinder head;

a spark plug connector removably located within the tube assembly; and

a spark plug secured to the spark plug connector.

15. The four stroke internal combustion engine according to claim 14, further comprising:

a pedestal within the cylinder head,

wherein the tube assembly is sealingly secured to the pedestal.

16. The four stroke internal combustion engine according to claim 14, wherein the tube assembly is plastic.

17. The four stroke internal combustion engine according to claim 14, wherein the spark plug connector further comprises:

a splash cover to prevent contaminants from entering the spark plug assembly.

18. The four stroke internal combustion engine according to claim 2, wherein at least one of the cam shaft and the support axle is offset with respect to the longitudinal axis.

19. The four stroke internal combustion engine according to claim 2, wherein the cam shaft is positioned closer to the exhaust valves than the intake valves.

20. The four stroke internal combustion engine according to claim 2, wherein the cam shaft and the support axle are positioned closer to the exhaust valves than the intake valves.

21. The four stroke internal combustion engine according to claim 2, further comprising:

a spark plug assembly connected to the cylinder head, wherein the spark plug assembly comprises

a tube assembly secured to the cylinder head;

a spark plug connector removably located within the tube assembly; and

a spark plug secured to the spark plug connector.

22. The four stroke internal combustion engine according to claim 21, further comprising:

a pedestal within the cylinder head, and

wherein the tube assembly is sealingly secured to the tube assembly.

23. The four stroke internal combustion engine according to claim 21, wherein the spark plug connector further comprises:

a splash cover to prevent contaminants from entering the spark plug assembly.

24. The four stroke internal combustion engine according to claim 21, wherein the spark plug connector further comprises:

a sealing assembly that forms a seal between the tube assembly and the spark plug connector.

25. A personal watercraft for at least one passenger, comprising:

a hull;

a seating assembly positioned on the hull adapted to accommodate at least one passenger; and

an internal combustion engine located within the hull, wherein the engine comprises

a crankcase secured to the hull having a crank shaft rotatably mounted therein;

a cylinder head connected to the crankcase, wherein the crankcase and the cylinder head form at least one cylinder;

at least one intake valve for the at least one cylinder, wherein the at least one intake valve has an intake valve axis;

at least one exhaust valve for the at least one cylinder, wherein the at least one exhaust valve has an exhaust valve axis;

a valve actuation assembly located in the cylinder head for operating the at least one intake valve and the at least one exhaust valve, wherein the valve actuation assembly is located substantially between the intake valve axis and the exhaust valve axis, the valve actuation assembly comprising

a cam shaft rotatably mounted within the cylinder head,

a support axle mounted within the cylinder head, offset from the cam shaft;

at least one exhaust rocker arm pivotally mounted on the support axle,

at least one intake rocker arm pivotally mounted on the support axle,

wherein the cam shaft is operatively coupled to the crankshaft such that rotational movement of the crankshaft is transferred to the cam shaft,

wherein the at least one exhaust rocker arm is operatively coupled to the cam shaft for operating the at least one exhaust valve, and

wherein the at least one intake rocker arm is operatively coupled to the cam shaft for operating the at least one intake valve; and

a lubrication system for lubricating the engine,

wherein the lubrication system includes a supply of lubricant to the cylinder head,

wherein the support axle has a central passageway extending therethrough, and

wherein at least a portion of the supply of lubricant flows through the central passageway in the support axle.

26. The personal watercraft according to claim 25, wherein the crankshaft and the cam shaft extend generally parallel to a longitudinal axis of the personal watercraft.

27. The personal watercraft according to claim 25, wherein each cylinder has a longitudinal axis, wherein each of the at least one intake valve and the at least one exhaust valve is disposed at an angle with respect to the longitudinal axis.

28. The personal watercraft according to claim 27, wherein the engine includes a pair of exhaust valves for each cylinder and the valve actuation assembly includes an exhaust rocker arm for each exhaust valve.

29. The personal watercraft according to claim 28, wherein each exhaust rocker arm comprises:

a cam follower located on one end of the exhaust rocker arm, wherein the cam follower is adapted to follow a profile of an exhaust cam lobe located on the camshaft; and

an exhaust hydraulic adjuster located on an opposite end of the exhaust rocker arm, wherein the exhaust hydraulic adjuster is adapted to contact and operate the exhaust valve in response to movement of the exhaust rocker arm by the exhaust cam lobe.

30. The personal watercraft according to claim **29**, wherein the exhaust hydraulic adjuster comprises:

an exhaust piston actuator slidably received within a cavity in the end of the exhaust rocker arm; and

a fluid passageway extending through the exhaust rocker arm to the support axle,

wherein the central passageway in the support axle is in fluidic communication with the fluid passageway such that fluid within the central passageway flows through the fluid passageway to the cavity to bias the exhaust piston actuator into engagement with the exhaust valve.

31. The personal watercraft according to claim **27**, wherein the engine includes a pair of intake valves for each cylinder and the valve actuation assembly includes a forked intake rocker arm for the pair of intake valves.

32. The personal watercraft according to claim **31**, wherein the forked intake rocker arm comprises:

a cam follower located on one end of the intake rocker arm, wherein the cam follower is adapted to follow a profile of an intake cam lobe located on the camshaft;

a first actuator arm for operating a first of the pair of intake valves;

a second actuator arm for operating a second of the pair of intake valves;

a pair of intake hydraulic adjusters located on an opposite end of the forked intake rocker arm,

wherein the intake hydraulic adjuster is adapted to contact and operate the pair of intake valves in response to movement of the intake rocker arm by the intake cam lobe.

33. The personal watercraft according to claim **32**, wherein each of the intake hydraulic adjusters comprises:

an intake piston actuator slidably received within a cavity in an end of the actuator arm, and a fluid passageway extending through the actuator arm to the support axle, and

wherein the central passageway in the support axle is in fluidic communication with the fluid passageway such that fluid within the central passageway flows through the fluid passageway to the cavity to bias the intake piston actuator into engagement with the intake valve.

34. The personal watercraft according to claim **33**, wherein the exhaust rocker arms are rotatably mounted on the support axle on opposite sides of the intake rocker arm.

35. The personal watercraft according to claim **34**, wherein each exhaust rocker arm comprises:

a cam follower located on one end of the exhaust rocker arm,

wherein the cam follower is adapted to follow a profile of an exhaust cam lobe located on the camshaft; and

an exhaust hydraulic adjuster located on an opposite end of the exhaust rocker arm, wherein the exhaust hydraulic adjuster is adapted to contact and operate the exhaust valve in response to movement of the exhaust rocker arm by the exhaust cam lobe.

36. The personal watercraft according to claim **35**, wherein the exhaust hydraulic adjuster comprises:

an exhaust valve actuator slidably received within a cavity in the end of the exhaust rocker arm; and

a fluid passageway extending through the exhaust rocker arm to the support axle,

wherein the central passageway in the support axle is in fluidic communication with the fluid passageway such that fluid within the central passageway flows through the fluid passageway to the cavity to bias the exhaust valve actuator into engagement with the exhaust valve.

37. The personal watercraft according to claim **33**, wherein the first actuator arm is spaced from the second actuator arm.

38. The personal watercraft according to claim **33**, further comprising:

a spark plug assembly disposed in the cylinder head, wherein the spark plug assembly is positioned between the first actuator arm and the second actuator arm.

39. The personal watercraft according to claim **38**, wherein the longitudinal axis of the spark plug assembly extends substantially parallel to the longitudinal axis of the intake valves.

40. The personal watercraft according to claim **38**, wherein the longitudinal axis of the spark plug assembly and the intake valve axis form an acute angle not greater than 20°.

41. The personal watercraft according to claim **38**, wherein the spark plug assembly comprises:

a tube assembly secured to the cylinder head;

a spark plug connector removably located within the tube assembly; and

a spark plug secured to the spark plug connector.

42. The personal watercraft according to claim **41**, further comprising:

a pedestal within the cylinder head, wherein the tube assembly is sealingly secured to the pedestal.

43. The personal watercraft according to claim **41**, wherein one end of the spark plug extends through a portion of the cylinder head into the cylinder.

44. The personal watercraft according to claim **29**, wherein at least one of the cam shaft and the support axle is offset with respect to the longitudinal axis.

45. The personal watercraft according to claim **44**, wherein the cam shaft is positioned closer to the exhaust valves than the intake valves.

46. The personal watercraft according to claim **44**, wherein the cam shaft and the support axle are positioned closer to the exhaust valves than the intake valves.

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